

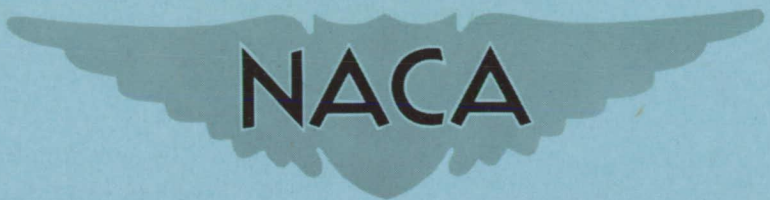
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RESEARCH MEMORANDUM

INVESTIGATIONS OF AIR-COOLED TURBINE ROTORS

FOR TURBOJET ENGINES

III - EXPERIMENTAL COOLING-AIR IMPELLER PERFORMANCE AND

TURBINE ROTOR TEMPERATURES IN MODIFIED J33 SPLIT-DISK

ROTOR UP TO SPEEDS OF 10,000 RPM

By Alfred J. Nachtigall, Charles F. Zalabak, and Robert R. Ziemer

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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SUMMARY

In an effort to reduce the extensive use of critical materials in the turbine rotor of gas turbine engines, an experimental investigation was conducted on a J33 turbojet engine modified for air cooling of the turbine disk and blades. Initial investigations into the general turbine disk cooling and structural characteristics of an experimental air-cooled split-disk rotor configuration have been reported previously as a part of this series.

The present report considers the cooling-air pumping characteristics of the internal impeller vane system between the parts of the split-disk rotor, the disk temperatures obtained after the engine configuration was modified to reduce the difference in temperature level of the two disks, the radial and tangential stress distribution based on extrapolated values of the disk temperature-difference ratio, the temperatures of the hollow tube-filled blades, and the heat transferred to the cooling air in the tail cone and the turbine disk.

Results of the investigation are presented for engine speeds up to 10,000 rpm (87 percent of rated speed) and for a range of cooling-air flow from approximately 0.02 to 0.10 of the engine combustion-gas flow. The results indicate a ratio of the static pressure in the region over the blade tip to the total pressure at the impeller inlet of 1.0 or above for cooling-air-flow ratios up to 0.035 over the range of corrected impeller speed investigated. The results also indicate that air bled from the compressor discharge would be at sufficient pressure to be introduced at the hub of the split disk and used to cool the blades over most of the range of cooling-air-flow ratio investigated. The difference in temperature level of the two disks was reduced at least 100° F by shutting off the external cooling air on the forward disk. Stress analysis of the two disks based on the new temperature distributions

indicated a substantial improvement over the stresses previously reported which were obtained with the more severe differences in temperature level. Indications were that substitution of noncritical alloys could be made for the disks of the rotor investigated. The heat transferred to the cooling air in the tail cone had no appreciable effect on the cooling of the disk and was approximately 8 percent of the estimated heat transferred to the blade. The heat transferred to the disk was about 3 percent of that estimated for the blade. The wide variation in the blade temperatures measured near the midchord of three instrumented blades was attributed to variations in the brazing bond between the internal tubes and the hollow blade shell.

INTRODUCTION

In an effort to reduce the extensive use of critical metals and alloys in the turbine rotor of gas turbine engines, an experimental investigation was conducted at the NACA Lewis laboratory on a J33 turbo-jet engine modified for air cooling of the turbine blades and disk. The investigation utilized a standard J33 disk which had been parted in the plane of rotation and had individual passages, for the introduction of cooling air to each blade, machined into the inside surfaces of both halves of the disk. The general aspects of the investigation, preliminary investigations up to 6000 rpm of disk cooling, and structural characteristics of this experimental air-cooled split-disk configuration are discussed in reference 1. The mechanical design of the turbine rotor and the tail cone, stress analysis, and burst test of the rotor configuration are presented in reference 2.

The introduction of cooling air through the disk to the blade base provided a strong cooling effect, but resulted in an unconventional disk configuration. In order to avoid large aerodynamic losses in the blade base and to enhance the pumping action, a system of vanes similar to those of a centrifugal impeller were provided integral with the rotor. The attempt to supply uniform chordwise distribution of the cooling air at the blade base resulted in a considerable increase in passage area from the impeller entrance to the tip. These large deviations from normal impeller design complicated the design of the disk structure at the rim, limited the aerodynamic performance potential of the impeller, and necessitated the experimental determination of the flow characteristics.

The need for additional engine modifications which would reduce the temperature level difference between the forward and the rear halves of the disk was indicated in reference 1. Following these modifications, the range of engine speed was extended.

The present report presents the results of the investigation over the extended speed range for the impeller cooling-air-flow characteristics as described by the cooling-air angle of incidence on the impeller vanes and the pressure ratio through the impeller. Further results are given on the disk temperatures. Stress levels in the disks at rated speed (11,500 rpm) are presented for the estimated temperature profiles obtained following the engine modifications which reduced the temperature level difference between the two halves of the disk. Those blade temperatures that aid in the evaluation of the uniformity of cooling-air distribution to the blades and the heat transferred to the cooling air in passing through the tail cone and through the disk are also presented.

Data are given for a range of engine speed from 4000 to 10,000 rpm for the impeller cooling-air-flow characteristics, 4000 to 8000 rpm for the disk temperatures, and 4000 to 6000 rpm for the blade temperatures. The cooling-air flow was varied from approximately 0.02 to 0.10 of the engine combustion-gas flow.

APPARATUS

Engine Modifications

The J33 turbojet engine used to conduct the investigation reported herein was the same as that used in reference 1. The modifications to the turbine disk and tail cone necessary to permit introduction of the cooling air to the base of the blades were described and discussed in reference 2. In figure 1 is shown the method of introducing cooling air from the laboratory high-pressure air system into the engine through four pipes located inside the modified tail-cone struts. The pipes terminated in a supply tube which introduced the cooling air into the split disk through a $3\frac{1}{4}$ -inch-diameter hole in the center of the rear disk.

The modified turbine rotor utilized a standard disk made of Timken alloy 16-25-6 that had been split in the plane of rotation and had individual passages to each blade machined into the inside surface of each half of the disk to form the internal impeller vane system shown in figure 2. The detailed geometry and dimensions of the split-disk impeller configuration can be seen in figure 3. The impeller vanes bring the cooling air from the rotor hub to the base of the blades and impart the tangential velocity of the rotor to the cooling air.

The internally cooled nontwisted rotor blades consisted of a hollow blade shell and blade base cast integrally of X-40, a high-temperature alloy. Figure 4 is a composite of three views of one of the blades used in the investigation. The coolant passage within the aerodynamic section was of constant cross section. The nine tubes, which terminated at

the blade platform, were inserted in the hollow shell and brazed in place (see fig. 4(a) for tube size and arrangement). Because the axial length of the passage within the blade was greater than the passage length in the serrated region of the rotor, a transition between the two cross sections was necessary and was accomplished within the blade base as indicated by the dashed lines in figure 4(c). The cross section of the cooling-air passage at the entrance to the blade base is shown in figure 4(b).

The modification recommended in reference 1 was made in order to more nearly equalize the temperature levels of the two halves of the turbine disk to permit operation at turbine speeds greater than 6000 rpm. The modification consisted of closing off the passages provided for the flow of ambient air for cooling the forward face of the disk in the original engine design. These passages were closed off at points A and B as shown in figure 5.

Instrumentation

Engine instrumentation. - Instrumentation for obtaining data to calculate engine mass flow, cooling-air mass flow, and effective gas temperature was the same as that used in the investigation described in reference 1. Static-pressure taps were located in the outer engine tail-cone casing directly over the cooling-air passage in the blades (see fig. 5). These taps were used in the determination of the static pressure of the combustion gas in the region of the cooling air discharge.

Rotating thermocouple system. - Two sets of 12 thermocouples were installed on the turbine rotor, as mentioned in reference 1. Described therein are the lead system, the thermocouple pickup, and the operation of selecting between the sets. One set of 12 rotating thermocouples was used for measuring the temperature on the two halves of the split disk. The locations of the disk thermocouples are shown in figure 6.

The other set of 12 rotating thermocouples was installed in groups of four on three of the 54 air-cooled rotor blades at intervals of approximately 120° . The locations of the four thermocouples as installed on each of the three instrumented blades are shown in figure 7. Three of the thermocouples on each of the instrumented blades were located at approximately the $1/3$ -span position - one near the leading edge, one in the midchord region on the pressure surface, and one near the trailing edge. The $1/3$ -span position was chosen because it is the critical region from the standpoint of allowable temperature and stress. The fourth thermocouple in each group was located in the cooling-air passage in the base of each instrumented blade to measure the temperature of the cooling air as it entered the blade (see fig. 7). The construction and the method of installation of these thermocouples are described in reference 3.

Cooling-air instrumentation. - The control of the flow of cooling air to the air-cooled turbine and the instrumentation necessary for calculation of flow rate were the same as those described in reference 1. The total temperature of the cooling air before entering the tail cone was taken as the temperature measured immediately downstream of the measuring orifice. Effective temperatures and total pressures of the cooling air at the hub of the disk were measured by means of a rake located in the cooling-air supply tube as shown in figure 5. The static pressures were measured by means of wall taps in the cooling-air supply tube (see fig. 5). The velocity determined by the total and static pressures enabled calculation of the total and static temperatures using the effective air temperatures and a thermocouple recovery factor.

EXPERIMENTAL PROCEDURES

The experimental procedure was the same as that described in reference 1. The engine was operated at various speeds and at each speed the cooling-air weight flow was varied from approximately 10 percent to 2 percent of the engine combustion-gas weight flow. A summary of the engine operating conditions is presented in table I. Series I and II data for 4000 and 6000 rpm, respectively, are data obtained while measuring temperatures of the blades and of the cooling air at the blade base. Data for impeller flow characteristics and disk temperature data were obtained for the remaining series representing engine speeds of 4000 through 10,000 rpm. Data for the disk temperatures at 10,000 rpm are not presented herein because of malfunction of the rotating thermocouple system at this speed.

CALCULATIONS AND METHODS

Engine weight flow, cooling-air weight flow, and effective gas temperature were calculated from engine instrumentation data by methods discussed in references 1 and 3.

Impeller Flow Characteristics

Angle of incidence. - In order to calculate the angle of incidence of the cooling air with the impeller vanes, it was necessary to determine the relative velocity of the cooling air in the split disk at the radius where the vanes begin. This relative velocity was determined from a vector summation of the radial velocity of the cooling air in the passage between the two halves of the split disk and the tangential velocity of the vanes. The tangential velocity of the vanes was calculated from the measured engine speed and the radius where the vanes

begin. Any tangential velocity that the cooling air may have acquired through viscous shearing forces between the air and the sides of the coolant passage below the vanes was neglected because calculations had indicated that this velocity would not be more than 6 percent of the disk tangential velocity. Inasmuch as no temperature and pressure measurements of the cooling air were made at this radius in the disk, the total pressures and total temperatures as obtained in the supply tube were assumed to remain constant up to the radius where the vanes begin. The static temperatures and pressures were then calculated and used to determine the radial air velocities at the vane inlet. The air-flow area at the vane inlet was taken as the geometric area since the blockage due to boundary layer was indeterminate in these investigations.

Pressure change. - The pressure change in the cooling-air impeller, which includes the passages within the blades, is presented herein as a function of the independent variables corrected weight flow $w_a \sqrt{\theta_{a,H}} / \delta_{a,H}$ and corrected impeller speed $N / \sqrt{\theta_{a,H}}$. (All symbols used herein are defined in the appendix.) Dimensional analysis shows these to be the two primary independent variables which describe the pressure change within a centrifugal impeller neglecting heat transferred to or from the working fluid, the Reynolds number, and the Froude number. Such a dimensional analysis is presented in reference 4.

It is realized that in the impeller considered herein more heat is transferred to the working fluid than would be encountered in any normal centrifugal impeller. Reference 5 presents an analysis in which the spanwise static-pressure variation in the coolant passage of an air-cooled blade can be evaluated by considering the simultaneous effect of the variation of the cross-sectional flow area, friction, rotational speed, and heat transfer to the cooling air. The available instrumentation, however, did not permit the evaluation of the temperature of the cooling air within the cooling-air passage at the blade tip or the pressure of the cooling air at the blade base, so that neither by use of the analytical method of reference 5 nor from experimental data could the pressure change be determined through the blade alone, which is the primary source of heat addition to the cooling air. Consequently, the cooling-air pressure change through the blade was included in the pressure change through the impeller.

As an approximation, the independent variables usually used for centrifugal impellers were used in presenting this pressure-change data. The data, as presented herein, are reliable for the evaluation of general trends and pressure levels for the configuration considered and for the conditions of operation. Care should be taken, however, in the extrapolation of these data to other conditions, such as higher altitudes, because the effect of heat addition has been neglected.

The pressure change across the impeller was taken as the ratio of the static pressure of the combustion gas in the region over the blade tip to the total pressure of the cooling air at the rotor hub. The static pressure of the combustion gas at the blade tip is a measure of the pressure against which the cooling air must be discharged. The effect of the cooling-air velocity head on the static tap in the tail-cone casing was neglected because the tangential component of the cooling air absolute velocity was large compared with the estimated radial velocity of the cooling air as it left the blade tip. In addition, the length of time during which a blade was opposite this tap was small as compared with the time during which no blade was opposite the tap, and the manometer system used to measure this pressure was insensitive to sudden short-time pressure fluctuations. The cooling-air weight flow w_a and the measured engine speed N were corrected to the conditions at the impeller inlet. That is,

$$\theta_{a,H} = T'_{a,H}/T_0$$

and

$$\delta_{a,H} = p'_{a,H}/p_0$$

where

$$T_0 = 518.4^\circ \text{ R}$$

$$p_0 = 2117 \text{ (lb/sq ft absolute)}$$

Compressor bleed air for cooling. - A convenient source of supply of cooling air for the turbine rotor is the engine compressor. The ability of the compressor to supply various quantities of cooling air was investigated by comparing the pressure available at the discharge of the compressor with the pressure required at the inlet to the cooling-air impeller to force the cooling air through the disk and blades. It was assumed that the cooling air was supplied from the discharge of the compressor and that the total pressure available at the inlet to the cooling-air impeller was equal to the static pressure at the compressor discharge. This means that all the velocity head available at the compressor discharge is assumed lost because of the method of bleeding and of the neglect of the pressure drop in the line from the point of bleedoff to the impeller hub. The total temperature at the inlet to the cooling-air impeller was assumed equal to the total temperature at the discharge of the compressor plus the measured total

temperature rise of the cooling air through the tail cone. Thus for a given engine speed and cooling-air weight flow from the measured data, an equivalent impeller speed and equivalent cooling-air weight flow can be obtained using this new cooling-air total temperature and an assumed value of required cooling-air total pressure. With the static- to total-pressure ratio of the impeller, evaluated from the impeller performance curves, and the measured static pressure at the tip of the blades known, the correct value of required cooling-air total pressure can be evaluated by successive approximations, and the ratio of available total pressure to required total pressure at the cooling-air impeller inlet can be determined.

Disk Temperature Correlations

Correlation methods. - The disk temperatures were correlated on the basis of values of the temperature-difference ratio $1-\phi$ which were plotted against the cooling-air-flow ratio w_a/w_g for the various engine speeds. The values of $1-\phi$ are presented in table II for engine speeds of 4000, 5000, 6000, and 8000 rpm. The basis for using such a correlation procedure and typical results are shown in reference 1. The effective cooling-air temperature $T_{a,e,H}$ used in calculating ϕ is that obtained from the rake in the cooling-air supply tube, and T_p is the disk metal temperature.

Estimated disk temperature profiles at rated engine conditions and cooling-air-flow ratio of 0.02. - The structural soundness of the disk configuration investigated must be evaluated at rated engine speed because of the expected difference in temperature level between the two disks. The disk temperature profiles were necessary in the plastic stress analysis used for this evaluation.

The disk temperature levels at rated engine speed (11,500 rpm) and for a cooling-air-flow ratio of 0.02 were determined from values of the temperature difference ratio $1-\phi$, the effective gas temperature, and the effective cooling-air temperature at rated engine conditions. Curves of $1-\phi$ against cooling-air-flow ratio, for each engine speed investigated and each of the 11 disk thermocouples, were extended to a cooling-air-flow ratio of 0.02, the flow at which the values of $1-\phi$ necessary to plot curves of $1-\phi$ against engine speed were taken. Extrapolating these curves to 11,500 rpm determined the required values of $1-\phi$.

Plastic Stress Analysis at 11,500 rpm

Plastic stress calculations were made following the methods outlined in reference 2 to determine whether, with the estimated disk temperature profiles determined by the methods presented in the preceding

section, the split-disk configuration considered herein could be safely operated at 11,500 rpm. The disk temperature profiles used were those obtained with the modifications incorporated in the engine configuration for equalizing the temperature level difference between the two disks. The disk stresses were calculated on the assumption of a rigid attachment between the disks at the rim as a result of the tightly fitting blade bases in the disk serrations.

Heat Transferred to Cooling Air

Heat transferred to cooling air in tail cone. - The temperature of the cooling air at the inlet to the turbine rotor has a direct effect on the cooling effectiveness of the disk and of the blades. As a consequence it is necessary to obtain an idea of the magnitude of the heat transferred to the cooling air as it passes through the high temperature regions of the tail cone. The heat-transfer rate to the cooling air in passing through the tail cone was calculated as the product of the total temperature rise from the inlet to the tail cone to the rotor hub, the cooling-air weight flow, and an average value of specific heat at constant pressure.

The heat picked up by the cooling air in the tail cone followed a heat path which is best described by reference to figure 1. The heat flows from the hot combustion gas to the modified strut and the tail-cone bullet, from the modified strut and tail-cone bullet to the gases between the modified strut and the tail-cone bullet and the cooling-air supply pipes and tube, from these gases to the cooling-air supply pipes and tube, and from the pipes and tube to the cooling air. An analysis of this heat flow can be made from which the temperature rise of the air in the tail cone could be calculated for conditions other than those investigated. However, to have some proof of the analysis would require measurements of the temperatures of the tail-cone bullet, modified strut, and cooling-air pipes, as well as of the flow rate of the gases between the tail-cone bullet and the cooling-air tube. The results of such an analysis are directly dependent on this weight flow rate of the gases in the tail-cone bullet. If the gases are relatively stagnant, the heat is transferred to the cooling-air tube by conduction; whereas if the flow rate is appreciable, the transfer is by convection. The structural requirements of the tail-cone configuration used necessitated two small holes in the tail-cone baffle plate (fig. 1) to relieve any pressure build-up within the tail-cone bullet. These holes and the clearances between the tail-cone bullet and the four cooling-air pipes undoubtedly permitted some flow of hot gases in the tail-cone bullet. To avoid this flow of gases in future designs would be desirable, since a stagnant air chamber around the cooling-air tube would reduce the amount of heat transferred to the cooling air. Inasmuch as the primary purposes of the investigation centered around the disk and blades, no

such instrumentation was installed. As a consequence, no detailed analysis of the heat flow is presented herein and the data concerning the heat transferred to the cooling air are presented only as a guide to determine whether or not it was appreciable. Since the data presented are for a given configuration and the tests were conducted at sea-level static conditions, care must be exercised in extrapolating these data.

Heat transferred to cooling air in turbine disk. - To determine the effect of the heat transfer in the disk on the operating temperatures of air-cooled turbine rotor blades, the heat-transfer rate to the cooling air while passing through the split-disk was calculated from measurements obtained during the investigations for engine speeds of 4000 and 6000 rpm. The temperature rises were obtained only at these speeds because of the limited speed range over which the engine was operated for the purpose of measuring the cooling-air temperature in the blade base and the blade wall temperatures. The heat-transfer rate was calculated as the product of the cooling-air-flow rate, an average value of c_p , and the temperature rise, due to heat transfer, of the cooling air during its passage from the hub of the rotor to the blade base. The temperature rise of the cooling air due to heat transfer was evaluated as follows:

$$\Delta T_{HT} = (T_{a,e,h} - T'_{a,H}) - (\omega^2 r_h^2 / 2Jg c_p)$$

The term $(\omega^2 r_h^2 / 2Jg c_p)$ accounts for the temperature rise due to the energy imparted to the cooling air in bringing it from the rotor hub to the blade base.

An analysis of the heat-transfer rate to the cooling air could be made considering the combustion gas as the heat source. Heat would be transferred to the disk by convection on the outside faces of the forward and rear disks, by radiation from members adjacent to the disk faces, and by conduction from the rim into the disks. The gas temperature in contact with the rear turbine faces would be modified by a leakage of cooling air through the seals around the cooling-air supply tube (fig. 1). Heat transferred to the disks would be conducted through the disks and transferred to the cooling air by convection. The analysis would enable the calculation of temperature rises of the cooling air through the disks at conditions other than those encountered during the investigation reported herein. The quantities other than those obtained during the investigation reported herein necessary for proof of such an analysis are temperatures of the gases in contact with the outside faces of the forward and rear disks, the temperature of members adjacent to the disk faces, and the quantity of leakage air. Care should be taken in applying the experimental results of this investigation to other operating conditions inasmuch as the curves presented are for a very limited speed range and are not substantiated by analysis.

RESULTS AND DISCUSSION

Impeller Cooling-Air-Flow Characteristics

The flow characteristics of the cooling-air impeller are discussed in the following paragraphs on the basis of the calculated angles of incidence of the cooling air with respect to the impeller vanes, the pressure ratios across the impeller, and the ratio of the available pressure at the compressor discharge to the required cooling-air pressure at the impeller hub.

Angle of incidence of cooling air with impeller vanes. - The angles of incidence of the flow of cooling air with respect to the impeller vanes are presented in order to give an indication of the flow existing in the impeller. The angle of the passage between the impeller vanes at the inlet, shown in figure 3 to be 30° from a radial line, was designed for a volume flow of 25 cubic feet per second at the rated engine speed of 11,500 rpm. Operation over a wide range of cooling-air flows at a constant engine speed produced a wide variation in the cooling-air angle of incidence with respect to the vanes, as shown in figure 8. Presented in the figure are the angles of incidence of the cooling air with respect to the vanes plotted as a function of the cooling-air flow for engine speeds of 4000, 6000, 8000, and 10,000 rpm. The angle of incidence on the vanes is positive at low cooling-air flows and becomes negative for high cooling-air flows for all the speeds except 10,000 rpm. For speeds of 8000 and 10,000 rpm, the angle of incidence at high flows tends to approach a terminal value, the attainment of which indicates that a limiting radial velocity at the vane entrance is encountered as a result of choking.

When the volume flow at the vane inlet was calculated it was noted that the design flow of 25 cubic feet per second occurred at approximately zero angle of incidence for the engine speed of 8000 rpm. At 10,000 rpm the design volume flow occurred at an angle of incidence of approximately 6° and the trend is toward a more positive angle at the design speed of 11,500 rpm. The angles presented in figure 8, as discussed previously, were calculated with the aid of the radial air velocities based on the geometric flow area at the vane inlet. The assumption of a partial flow-area blockage would result in the calculation of higher radial velocities than those based on the geometric area. The calculated angles of incidence would therefore be more negative than indicated and the design volume flow at design speed would occur at an angle less positive than would be indicated.

Impeller pressure ratio. - The pressure ratio across the impeller is of interest because it determines the cooling-air pressure required at the rotor hub for a given combination of cooling-air flow and engine

operating conditions. The development of a pressure rise across the impeller limits the degree of precompression necessary for the required coolant flow and results in lower cooling-air temperatures, thereby enhancing the cooling of the rotor. This particular impeller configuration developed a pressure rise over all impeller speeds investigated for cooling-air-flow ratios up to approximately 0.035, as is indicated in figure 9. Plotted here are the ratio of static pressure in the region over the blade tips, into which the cooling air discharged, to the total pressure of the cooling air at the impeller hub against corrected cooling-air weight flows for corrected impeller speeds of 4000, 6000, 8000, and 10,000 rpm. It can be seen that for each speed the highest pressure ratios were obtained for the lowest cooling-air flows. As cooling-air flow increases for any given speed, the pressure ratio developed decreases. This decrease in pressure ratio as cooling-air flow increases can be partially explained by referring to figure 8. At low speeds, the decrease in pressure ratio is due in part to the pressure losses associated with the high negative angles of incidence of the cooling air with the impeller vanes. As the engine speed was increased, however, the angles of incidence became more favorable over the range of cooling-air weight flow investigated. At 10,000 rpm the angle of incidence is positive over the entire cooling-air weight flow range. The decrease in pressure ratio with increasing cooling-air flow (fig. 9) must be due to choking in the cooling-air-flow passages. In all cases, pressure ratios of 1.0 or greater occurred at positive angles of incidence over the impeller speed range investigated.

Over the entire range of impeller speed investigated, pressure ratios of 1.0 or above were obtained for a cooling-air-flow ratio of 0.035. This condition probably holds true up to rated engine speed. Cooling-air-flow ratios in the vicinity of 0.03 are currently considered practical in the application of air-cooled turbine rotor blades, although efforts are being directed toward the development of blades which require even smaller cooling-air-flow ratios.

Cooling-air impellers built into the disk of air-cooled turbines should be such that they are capable of overcoming the internal pressure losses. In some cases, if rotor blades of high cooling effectiveness which permit low cooling-air flows are used, some pressure rise will be possible. Because the pressure ratios are only slightly greater than 1.0, some means of mechanical precompression of the cooling air before it is introduced at the hub of the rotor will be required for all cases in which the engine operating conditions impose pressures at the blade tip higher than ram inlet.

Comparison of compressor discharge pressure and required cooling-air pressure at impeller hub. - When air cooling of turbine blades is applied to airborne gas turbine engine installations, one means of supplying the cooling air is to bleed it from the engine compressor.

For the series of investigations conducted, figure 10 is presented to show the range of cooling-air-flow ratio over which compressor discharge air is at sufficient pressure to be introduced at the hub of this particular split-disk rotor. The pressure drop in the lines from the bleed-off point to the rotor hub has been neglected; however, the pressure losses will be large in an actual engine installation unless the system is carefully designed. The ratio of compressor discharge static pressure to required cooling-air total pressure at the rotor hub is plotted against the cooling-air-flow ratio for engine speeds of 4000, 6000, 8000, and 10,000 rpm. The static pressure at the compressor discharge is higher than that required at the impeller hub for the four engine speeds presented over the greater part of the range of cooling-air-flow ratio investigated. The results as presented in figure 10 show general levels and trends as these results are based on the assumption that the pressure ratio across the impeller can be described by the corrected cooling-air weight flow and the corrected impeller speed as shown in figure 9. The effect of heat transfer to the cooling air on the pressure ratio has been disregarded.

Cooling air bled from the compressor discharge would be at a considerably higher temperature than that of the high-pressure laboratory air supplied to the engine in the test cell, and higher cooling-air-flow ratios would be required to maintain a given operating blade temperature. Because of the apparent excess pressure available over most of the range of cooling-air-flow ratio, some pressure drop would be allowed for an intercooler and the bleed-off system ducting. Installation of the intercooler would lower the cooling-air temperature and therefore reduce the required cooling-air flow. The heat from the cooling air could be exchanged to the fuel or the ram air during flight.

Disk Temperatures

A correlation of the disk temperature-difference ratio with engine speed, obtained following the engine modification made in an attempt to equalize the temperature profiles in the two halves of the disk, is presented herein. Also shown are the estimated disk temperature profiles for rated engine speed based on the extrapolation of this correlation to rated engine speed.

Disk temperature correlation. - The variations with engine speed of the temperature parameters $1-\varphi$ for the disk thermocouple locations 1 through 11 are presented in figure 11 for a cooling-air-flow ratio of 0.02. The values of $1-\varphi$ for 4000, 5000, 6000, and 8000 rpm over the range of cooling-air-flow ratio investigated are presented in table II for which the engine operating conditions are summarized in table I. The temperature parameter $1-\varphi$ for the thermocouples located near the disk rim (6, 7, 8, and 9) decreased with engine speed (fig. 11).

The values of $1-\Phi$ for the remaining thermocouple locations remained approximately constant over the speed range investigated. The decrease of $1-\Phi$ with engine speed for the thermocouple locations near the disk rim means that the disk temperature is not increasing so rapidly as is the effective gas temperature if the cooling-air inlet temperature is assumed to remain constant. This effect is probably caused by an increase in the rate of heat transfer from the disk to the cooling air in this region. Since a complete theoretical analysis of all the factors affecting the disk temperatures is unavailable at present, the exact cause for this increase in heat-transfer rate cannot be determined. It can also be noted that at 4000 rpm the range of $1-\Phi$ for thermocouple locations 6 in the forward disk and 8 in the rear disk is 0.38 to 0.79, while at 8000 rpm the range has been decreased to 0.31 to 0.51. That is, as the engine speed is increased the temperature difference between the forward and rear disks decreased in the rim section of the configuration investigated herein.

Estimated disk temperature profiles at rated engine speed. - A comparison of the estimated disk temperature profiles, obtained by extrapolating to rated engine speed the plots of $1-\Phi$ against engine speed as was done in figure 11, and the estimated disk temperature profiles as presented in reference 1 are shown in figure 12. Eliminating the flow of cooling air on the external face of the forward disk was beneficial in reducing the difference in temperature level between the two disks. At a disk radius of 3 inches the temperature of the forward disk increased approximately 150°F , while the temperature in the rear disk increased only 50°F . At a disk radius of 7 inches the temperature of the forward disk again increased approximately 150°F , while the temperature of the rear disk remained about the same. Thus, the difference in temperature level of the two disks was reduced between 100° and 150°F by closing off the external cooling air. Also, the sharp temperature gradient near the rim of the forward disk, as would be obtained from the dashed line temperature profile (fig. 12), has been greatly reduced. The maximum disk temperature as presented in figure 12 is approximately 700°F for the data presented herein with a 120°F cooling-air inlet temperature. With a cooling-air inlet temperature of 445°F , as might be obtained when using compressor bleed air, the maximum disk temperature based on the extrapolated values of $1-\Phi$ would be slightly below 900°F . The substitution of noncritical materials in the turbine disks is believed to be satisfactory for operation of turbines at this temperature level.

Plastic Stress Analysis at 11,500 rpm

A stress analysis was made for the forward and rear disks based on the estimated disk temperature profiles shown in figure 12 for rated engine conditions. Since the rear disk tends to expand more than the

forward disk, a system of restraining forces was set up with the blade bases in the disk serrations transferring these forces from one disk to the other. An initial assumption was therefore made that the radial displacements of the rims of the two disks must be equal at any particular time. The graphical solution of the restraining load for the operating conditions at rated speed is shown in figure 13. The radial displacements of the rims of the two disks are seen to be equal at a restraining load of 43,600 pounds per square inch. For the forward disk, the restraining load adds to the normal rim loading whereas for the rear disk the restraining load is in the opposite direction, and subtracts from the normal rim loading. The normal rim loading due to the blades and the serrated portion of the disks is 21,700 pounds per square inch for each disk. The total effective rim loading for the forward disk is therefore 65,300 pounds per square inch, and for the rear disk, -22,000 pounds per square inch. These rim loadings were used to calculate the stresses in the two disks as shown in figure 14. Plastic flow was taken into account for the forward disk. In comparing these results with those of figure 18 in reference 2 it can be seen that the decrease in the temperature-level difference between the forward and rear disks was quite beneficial in reducing the stress levels of the two disks.

In comparing these results with those shown in reference 2 for a more severe temperature difference between the two disks, it is also well to consider the effect of probable differences in the radial displacements of the rims of the two disks as permitted by a given amount of looseness of the blade bases in the disk serrations. If a difference of 0.010 inch in the radial displacements of the two rims were permitted, then according to figure 13 the restraining load would be reduced from 43,600 pounds per square inch to approximately 20,000 pounds per square inch. The total effective rim loading for the forward disk would be 41,700 pounds per square inch and for the rear disk, 1,700 pounds per square inch. In reference 2, it is pointed out that a difference of 0.010 inch in the radial displacements of the rims of the disks for the condition of more severe temperature difference would only result in a reduction of the restraining load from 50,000 pounds per square inch to 41,000 pounds per square inch. This significant improvement as a result of the less severe temperature difference indicates that under actual operating conditions in which some difference in the radial displacements of the disk rims probably does occur, the stresses in the forward disk will be considerably lower than those shown in figure 14. Assuming equal elongation of the disks at the rim and using the method of calculation substantiated by burst test as presented in reference 2 show the configuration investigated herein to be structurally sound at rated engine conditions, provided the difference in temperature level of the two disks is approximately that estimated in this report. Allowing a difference of radial displacement of 0.10 inch between the rims results in even more conservative stress levels.

Blade Temperatures

The blade temperature profiles for each of the three instrumented blades are presented for a cooling-air-flow ratio of 0.021 and an engine speed of 6000 rpm in figure 15. The profiles presented are similar to those obtained over the range of cooling-air-flow ratio investigated at 4000, 5000, and 6000 rpm. In all cases the temperatures at the leading and trailing edges were higher than those obtained in the midchord region. The high temperatures at the leading edge are caused by the high, external heat-transfer coefficient at the leading edge and the unfavorable ratio of internal to external heat-transfer flow area. At the trailing edge, the high temperatures are the result of the long conduction path and the unfavorable ratio of internal to external heat-transfer flow area. The temperatures measured at the midchord varied considerably for the three blades. The temperature at the midchord of blade 1 was about 140° F hotter than that of blade 2 and about 110° F hotter than that of blade 3.

Preliminary examination showed that the midchord temperature of blade 1 was hotter because of a defective brazing bond between the tubes and the shell in the midchord region where the thermocouple was located. This trouble had been indicated by visual observation of oxidation patterns on the blades. The suggested lack of a good thermal brazing bond between the tubes and the shell was verified when blade 1 failed, with the tubes drawing out of the portion of the shell remaining on the blade base as shown in figure 16. Examination indicated that little or no brazing bond existed between the blade shell and the tubes in the midchord region on the pressure side where the thermocouple was located. The need for a uniform brazing bond and good thermal contact between the blade shell and the internal tubes is evident from this experience.

One of the purposes of this investigation was to obtain data from the instrumented blades that would enable the determination of the uniformity of the cooling-air distribution to the blades by comparing the temperatures at similar locations. Because of the wide temperature variations caused by the nonuniformity of the brazing bond, no definite conclusions could be reached with regard to the flow distribution. However, the elimination of blade 1 from the comparison on the grounds of poor brazing bond leaves blades 2 and 3, the temperatures of which vary no more than 35° at the same location on each blade, indicating that essentially equal coolant flows are supplied to each blade.

Heat Transferred to Cooling Air

Heat transferred to cooling air in tail cone. - The variation of the calculated heat-transfer rate to the cooling air with cooling-air-flow rate through the tail cone and tail-cone bullet is presented in

figure 17. The heat-transfer rate is nearly constant over the coolant-flow range for any one speed, and for practical purposes may be taken as 10 Btu per second for all speeds presented.

The effect of this constant heat input to the cooling air on the effectiveness of the disk cooling can best be illustrated by means of numerical examples at a low cooling-air-flow ratio. Table II summarizes calculated disk and blade temperatures for various heat-transfer rates to the cooling air. The conditions for the calculations are listed above the table. Consider the disk temperatures obtained when $1-\phi$ is 0.09 (see fig. 11, thermocouple 3). Reducing the heat transferred in the tail cone from 10 Btu/sec to 0 resulted in a decrease in disk temperature of only 26° F. For a higher value of $1-\phi$ of 0.41 (see fig. 11, thermocouple 8), the temperature decrease was even smaller. A system as used in the configuration reported herein for supplying the cooling air through the high-temperature regions of the tail cone under the limited conditions of these tests therefore appears to impose no practical penalty on the operation of the air-cooled turbine disks because of the temperature rise of the cooling air based on the heat-transfer rate as shown by figure 17.

Heat transferred to cooling air in disk. - The variation with cooling-air-flow rate of the heat-transfer rates to the cooling air during the passage of the air from the rotor hub to the blade base is shown in figure 18 for engine speeds of 4000 and 6000 rpm. The heat-transfer rate is low, in the range of 1.0 to 7.0 Btu per second, which is lower than the rate of heat transfer to the cooling air in the tail cone (see fig. 17). There is a variation of approximately 4.0 Btu per second at both speeds over the range of cooling-air flow investigated. For convenience, a single line has been faired through the data points presented. This line, however, is not intended to define the operating heat-transfer rate at conditions other than those for which the data were obtained.

In order to determine the practical significance of this variation of heat transfer to the cooling air (thereby increasing the cooling-air temperature at the blade base) on the operating temperatures of the blades, numerical illustrations are presented in table III: The faired curve (fig. 18) shows a variation of 2.5 to 6.0 Btu per second over the cooling-air-flow range presented, and for the purpose of illustration only it was assumed to represent the condition at rated engine speed (11,500 rpm). As in the preceding section the conditions for the calculations are those listed above the table. Values of 0.30 and 0.55 were chosen for the blade temperature-difference ratio

$$\frac{(T_{g,e} - T_B)}{(T_{g,e} - T_{a,e,h})}$$

For a heat-transfer rate of 10 Btu per second in the tail cone the maximum blade temperature increase is 5°F when the heat-transfer rate in the disk is varied from 2.5 to 6.0 Btu per second and is obtained for the blade temperature-difference ratio of 0.55. Therefore, for practical purposes, the heat-transfer rate to the cooling air during its passage from the rotor hub to the base of the blades may be assumed constant at a given speed when the variations of the rate are similar to those shown in figure 18.

The heat-transfer rates to the cooling air in the tail cone and in the disk were compared with an estimated heat-transfer rate in the air-cooled rotor blades at an engine speed of 6000 rpm and a cooling-air-flow ratio of 0.056 in order to obtain a knowledge of the relative magnitudes of these quantities. The comparison showed the rates of heat transfer to the cooling air in the tail cone and in the disk to be about 8 percent and 3 percent, respectively, of the rate of heat transfer in the blades.

General Discussion

The investigations conducted so far (references 1 and 2 and that shown herein) have shown from considerations of the disk temperatures and stresses the feasibility of operating a split-disk type of air-cooled turbine rotor at cooling-air-flow ratios which were adequate to cool the blades. Under these conditions the disk temperatures were low enough to indicate that direct substitution of noncritical alloys in the disk is permissible, although practical considerations such as corrosion protection must still be investigated. Also, the heat transferred to the cooling air before it enters the blades and the pressure required at the cooling-air impeller inlet were not excessive for the configuration tested. However, the uniformity of the distribution of cooling air to the blades by means of the impeller vanes must yet be evaluated by experiments, since a shortage of cooling air to one blade will mean higher required cooling-air-flow ratios. Methods of supplying the cooling air by bleeding the engine compressor and the design of low pressure drop ducting systems should be investigated. The effect of this bleeding, pumping the cooling air through the rotor, and discharging the cooling air into the combustion-gas stream on engine performance must be evaluated. Also, to expedite the design of air-cooled turbine disks, a method for predicting the disk temperature distributions for various engine configurations is required.

SUMMARY OF RESULTS AND CONCLUSIONS

Results of an investigation conducted on a J33 turbojet engine modified for air cooling of the turbine disk and blades have been presented for engine speeds up to 10,000 rpm which is 87 percent of rated

speed, and for a range of cooling-air flow from approximately 0.02 to 0.10 of the engine combustion-gas flow. Some of the results observed and the conclusions drawn from them are as follows:

1. Air bled from the compressor discharge would be at sufficient pressure to be introduced at the inlet of the cooling-air impeller and used to cool the blades for cooling-air-flow ratios over most of the flow range for the speeds investigated.

2. The ratio of the static pressure in the region over the blade tips to the total pressure of the cooling air at the inlet of the cooling-air impeller was approximately 1.0 for cooling-air-flow ratios up to 0.035 over the range of corrected impeller speed and cooling-air weight flow investigated.

3. A consideration of the angle of incidence of the cooling air with the vanes of the internal cooling-air impeller indicated that choking of the cooling-air flow occurs at engine speeds of 8000 and 10,000 rpm at high coolant flows.

4. The difference between the estimated temperature levels of the forward and rear disks at rated engine conditions was reduced 100° to 150° F by eliminating the external cooling air on the forward disk. With these reduced temperature-level differences the plastic stress distribution was more favorable and the configuration investigated was considered to be structurally sound at rated engine conditions. Under conditions of actual operation during which some radial displacement between the rims probably did occur, the stress levels were further and more readily reduced as a result of the reduction in temperature-level difference.

5. The substitution of noncritical alloys for the disks of the rotor investigated was indicated to be feasible. The maximum estimated temperature of the disks, based on the correlation methods presented, was below 900° F at rated engine conditions and a cooling-air-flow ratio of 0.02 even for cooling-air inlet temperatures of 445° F.

6. The wide variation in the blade temperatures measured near the midchord of the three instrumented blades was attributed to variations in the brazing bond between the tubes in the hollow blade shell and the inside walls of the shell. This variation of brazing bond also eliminated any definite conclusions as to the uniformity of the distribution of cooling air to the blades.

7. The heat transferred to the cooling air in the tail cone over the range of conditions investigated imposed no practical penalty on the cooling of the turbine disks. The heat transferred to the cooling air in the disk was practically constant over the speeds and cooling-air-flow ratios investigated. At 6000 rpm and a cooling-air-flow ratio of 0.056, the heat transferred to the cooling air in the tail cone and the disk was estimated to be approximately 8 percent and 3 percent, respectively, of that transferred to the blades.

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APPENDIX - SYMBOLS

The following symbols are used in this report:

c_p	specific heat at constant pressure, Btu/(lb)(°F)
g	ratio of absolute to gravitational unit of mass, lb/slug
J	mechanical equivalent of heat, 778.3 ft-lb/Btu
N	engine speed, rpm
p	static pressure, lb/sq ft absolute or in. Hg absolute
p'	total pressure, lb/sq ft absolute or in. Hg absolute
Q	heat-transfer rate, Btu/sec
r	radius, ft
T	temperature, °R or °F
T'	total temperature, °R or °F
w	weight flow rate, lb/sec
α	angle of incidence of cooling air with impeller vanes, deg
Δ	prefix to indicate change
$\delta_{a,H}$	pressure correction ratio for cooling air at rotor hub, $p'_{a,H}/p_0$
$\theta_{a,H}$	temperature correction ratio for cooling air at rotor hub, $T'_{a,H}/T_0$
ϕ	disk temperature-difference ratio, $(T_{g,e}-T_D)/(T_{g,e}-T_{a,e,H})$
ω	angular velocity of rotor, radians/sec

Subscripts:

a	cooling air
B	blade
c	compressor

D disk
e effective
f fuel
g combustion gas
H station at rotor hub
HT heat transfer
h station at blade base
T tip
O NACA sea-level air
1-12 refer to disk thermocouple locations

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TABLE I - SUMMARY OF ENGINE OPERATING CONDITIONS

Series	Run	Engine speed N (rpm)	Average compressor-inlet condition		Average fuel flow $\dot{w}_f$ (lb/sec)	Combustion-gas flow $\dot{w}_g$ (lb/sec)	Cooling-air flow $\dot{w}_a$ (lb/sec)	Coolant-flow ratio $\dot{w}_a/\dot{w}_g$	Calculated effective gas temperature $T_{g,e}$ (°F)	Cooling-air inlet total temperature (hub) $T'_{a,H}$ (°F)	Cooling-air inlet total pressure (hub) $p'_{a,H}$ (in. Hg abs)	Effective cooling-air temperature at blade base (average) $T_{a,e,h}$ (°F)	Service air entrance temperature (°F)
			Pressure p' (in. Hg abs)	Temperature T' (°F)									
I	1	3998	29.68	29.4	0.350	21.78	2.03	0.093	1150	68	38.30	79	43
	2	4006				22.13	1.87	.085	1142	67	36.95	80	44
	3	3998				22.02	1.80	.082	1143	69	36.22	80	42
	4	4008				22.30	1.67	.075	1136	70	35.10	79	42
	5	4010				22.44	1.57	.067	1132	71	34.00	84	42
	6	3998				22.34	1.33	.059	1126	78	32.50	91	46
	7	4000				22.61	1.21	.054	1116	81	31.70	94	47
	8	4004				22.56	1.12	.050	1112	83	31.22	99	47
	9	4004				22.57	.94	.042	1111	91	30.30	109	47
	10	4006				22.64	.69	.031	1103	102	29.38	137	50
II	1	5978	29.14	53.2	0.502	30.33	3.26	0.108	1144	76		87	43
	2	6000				30.93	2.88	.093	1123	64		90	43
	3	5986				31.17	2.62	.084	1110	68		91	44
	4	5980				31.53	2.34	.074	1101	71		92	44
	5	5990				31.85	2.08	.065	1099	70		96	44
	6	5992				32.23	1.80	.056	1099	76		102	45
	7	5998				32.43	1.55	.048	1099	80		107	46
	8	5988				32.40	1.30	.040	1096	90		122	48
	9	6000				32.31	.97	.030	1092	106		146	50
	10	5994				31.97	.68	.021	1087	124		185	54
III	1	3991	29.26	87.0	0.348	19.49	2.03	0.104	1315	92			71
	2	3976				19.56	1.78	.091	1287	94			71
	3	3991				19.80	1.56	.079	1269	96			71
	4	4010				19.86	1.30	.062	1259	101			71
	5	4007				19.88	1.05	.053	1258	106			71
	6	3995				19.98	.78	.039	1265	124			72

TABLE I - SUMMARY OF ENGINE OPERATING CONDITIONS - Concluded

Series	Run	Engine speed N (rpm)	Average compressor-inlet condition		Average fuel flow $\dot{w}_f$ (lb/sec)	Combustion-gas flow $\dot{w}_g$ (lb/sec)	Cooling-air flow $\dot{w}_a$ (lb/sec)	Coolant-flow ratio $\dot{w}_a/\dot{w}_g$	Calculated effective gas temperature $T_{g,e}$ (°F)	Cooling-air inlet total temperature $T_{a,H}$ (°F)	Cooling-air inlet total pressure $P_{a,H}$ (in. Hg abs)	Effective cooling-air temperature at blade base $T_{a,e,h}$ (°F)	Service air entrance temperature (°F)
			Pressure P_i (in. Hg abs)	Temperature T_i (°F)									
IV	1	4976	29.26	84.4	0.424	24.48	2.45	0.100	1250	91			68
	2	4994				24.77	2.16	.087	1228	92			67
	3	5000				25.05	1.78	.071	1209	93			67
	4	5000				25.29	1.48	.059	1203	96			67
	5	4988				25.26	1.14	.045	1214	109			68
	6	5009				25.43	.79	.031	1223	140			69
V	1	6000	29.33	88.1	0.495	29.76	3.03	0.102	1205	80	49.76		68
	2	6010				30.28	2.64	.087	1177	79	44.61		66
	3	6003				30.71	2.17	.071	1160	80	38.58		66
	4	6015				31.05	1.66	.053	1153	89	33.31		67
	5	5998				31.26	1.22	.039	1160	96	29.73		68
	6	6000				31.24	.77	.025	1162	115	27.36		69
VI	1	8008	29.29	96.1	0.704	39.60	4.38	0.105	1325	82	64.94		70
	2	7993				40.14	3.66	.091	1289	76	55.00		68
	3	8005				40.61	2.86	.071	1274	75	43.30		67
	4	7991				40.94	2.04	.050	1264	89	34.66		68
	5	8008				41.09	1.27	.031	1299	111	28.55		70
	6	8008				41.27	.94	.023	1292	131	26.60		71
VII	1	9955	29.17	76.4	1.007	54.08	5.69	0.105	1378	64	81.89		64
	2	9967				53.77	5.32	.099	1371	64	76.48		64
	3	9986				54.01	4.73	.088	1364	63	68.16		64
	4	9991				54.02	3.84	.071	1354	63	55.98		64
	5	9984				53.43	2.71	.051	1340	83	40.83		63
	6	9993				53.02	1.67	.031	1334	97	30.17		65

TABLE II - SUMMARY OF 1- ϕ FOR DISK

Series	Run	Nominal engine speed N (rpm)	Coolant-flow ratio w_a/w_g	Effective cooling-air temperature at rotor hub $T_{a,e,H}$ ($^{\circ}F$)	1- ϕ_1	1- ϕ_2	1- ϕ_3	1- ϕ_4	1- ϕ_5	1- ϕ_6	1- ϕ_7	1- ϕ_8	1- ϕ_9	1- ϕ_{10}	1- ϕ_{11}
III	1	4000	0.104	80	0.107	0.093	0.049	0.115	0.079	0.172	0.245	0.330	0.122	0.075	0.062
	2		.091	84	.111	.099	.053	.119	.087	.185	.259	.371	.140	.083	.072
	3		.079	88	.112	.103	.056	.131	.092	.206	.284	.423	.168	.093	.076
	4		.062	96	.118	.108	.053	.143	.101	.221	.309	.485	.187	.110	.092
	5		.053	104	.123	.112	.064	.159	.126	.256	.346	.554	.224	.128	.107
	6		.039	123	.131	.122	.075	.180	.151	.298	.393	.598	.281	.173	.151
IV	1	5000	0.100	78	0.102	0.090	0.039	0.104	0.069	0.154	0.233	0.272	0.114	0.067	0.053
	2		.087	81	.106	.092	.042	.113	.077	.169	.247	.300	.128	.076	.063
	3		.071	86	.110	.100	.049	.127	.090	.191	.273	.346	.158	.091	.075
	4		.059	92	.116	.105	.054	.134	.100	.213	.300	.406	.187	.107	.086
	5		.045	108	.122	.110	.061	.154	.120	.250	.345	.537	.247	.141	.119
	6		.031	140	.121	.113	.068	.179	.149	.301	.411	.549	.325	.232	.264
V	1	6000	0.102	72	0.102	0.094	0.040	0.107	0.072	0.146	0.221	0.204	0.110	0.070	0.054
	2		.087	71	.110	.099	.047	.114	.078	.157	.232	.232	.128	.083	.063
	3		.071	74	.110	.106	.053	.124	.088	.177	.252	.272	.155	.095	.077
	4		.053	84	.119	.110	.058	.138	.103	.206	.287	.354	.207	.126	.100
	5		.039	94	.128	.121	.070	.160	.127	.245	.336	.451	.280	.185	.185
	6		.025	114	.138	.135	.091	.198	.171	.311	.408	.537	.360	.272	.309
VI	1	8000	0.105	74	0.114	0.104	0.027	0.108	0.064	0.135	0.197	0.163	0.118	0.075	0.061
	2		.091	69	.131	.121	.048	.127	.081	.159	.221	.199	.146	.097	.085
	3		.071	68	.138	.131	.058	.143	.099	.188	.250	.272	.182	.125	.110
	4		.050	84	.146	.137	.067	.160	.117	.222	.287	.353	.231	.150	.127
	5		.031	108	.149	.144	.079	.183	.146	.266	.335	.437	.302	.226	.251
	6		.023	129	.153	.151	.089	.200	.167	.296	.370	.470	.335	.259	.307

TABLE III - SUMMARY OF CALCULATED DISK AND BLADE TEMPERATURES

FOR VARIATIONS OF HEAT-TRANSFER RATE TO COOLING AIR

$T_{g,e} = 1445^{\circ} \text{ F}$; air temperature at tail-cone inlet = 70° F ;
 $w_a/w_g = 0.02$; $w_g = 72 \text{ lb/sec}$; $c_p = 0.24$; $N = 11,500 \text{ rpm}$

$1-\phi$	$\frac{T_{g,e}-T_B}{T_{g,e}-T_{a,e,h}}$	Heat-transfer rate to cooling air (Btu/sec)		T_D ($^{\circ} \text{ F}$)	T_B ($^{\circ} \text{ F}$)
		In tail cone	In disk		
0.09		10	2.5	220	
.09		0	2.5	194	
.41		10	2.5	651	
.41		0	2.5	634	
	0.30	10	6.0		1067
	.30	10	2.5		1064
	.55	10	6.0		751
	.55	10	2.5		746

NACA

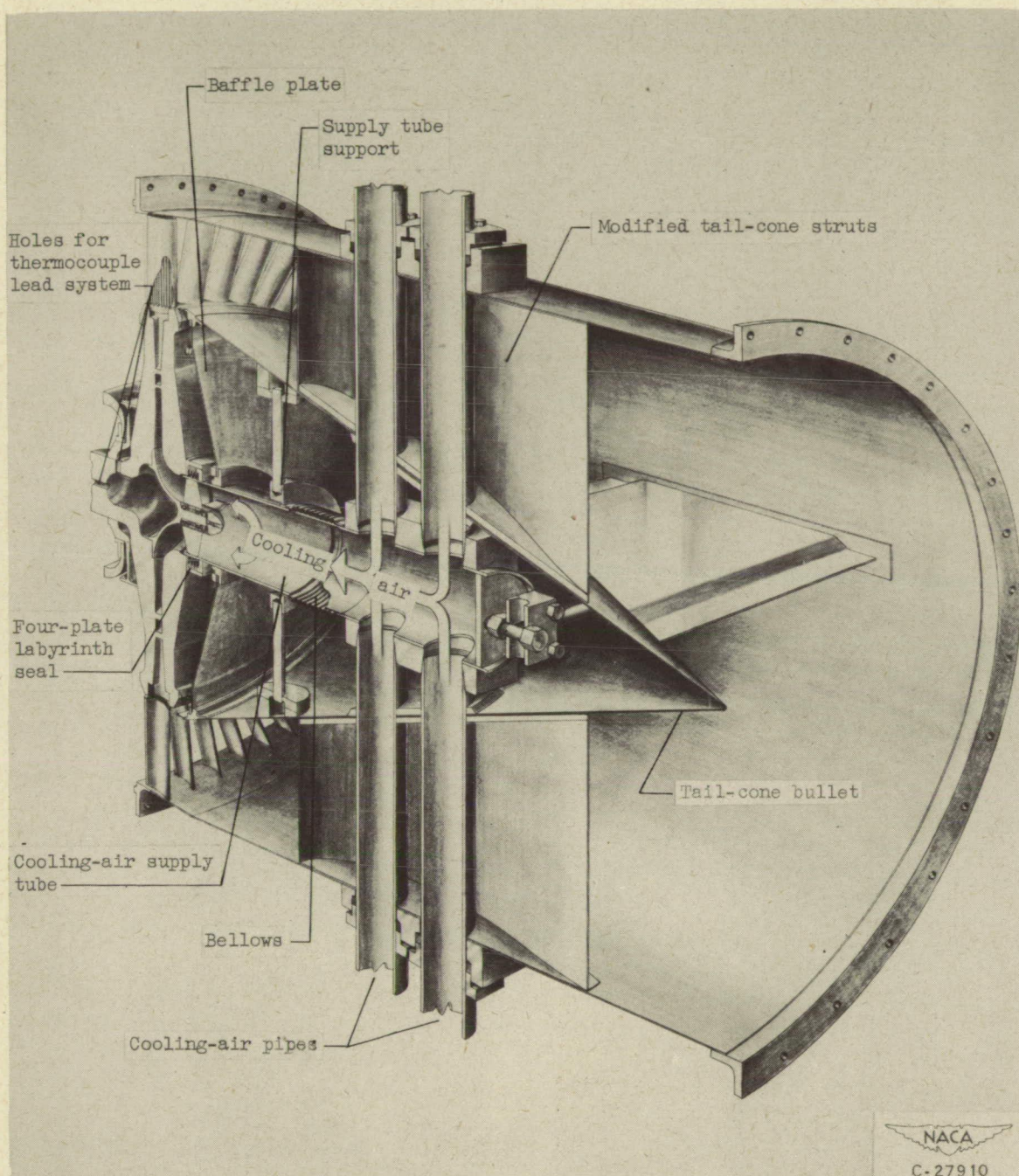
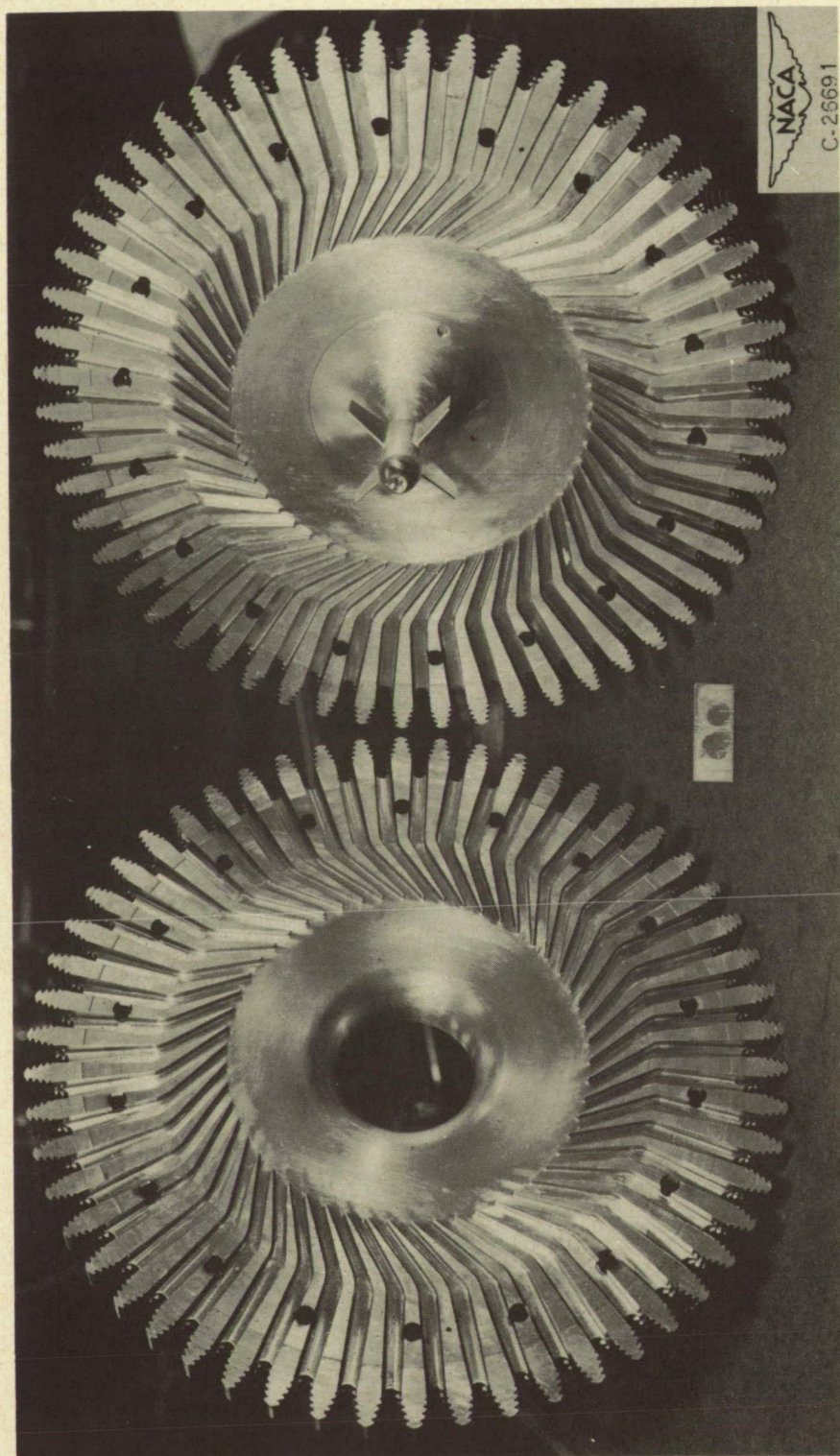


Figure 1. - Tail-cone modifications for introduction of cooling air.



Forward disk

Rear disk

Figure 2. - Internal cooling-air impeller vanes in both halves of wheel.

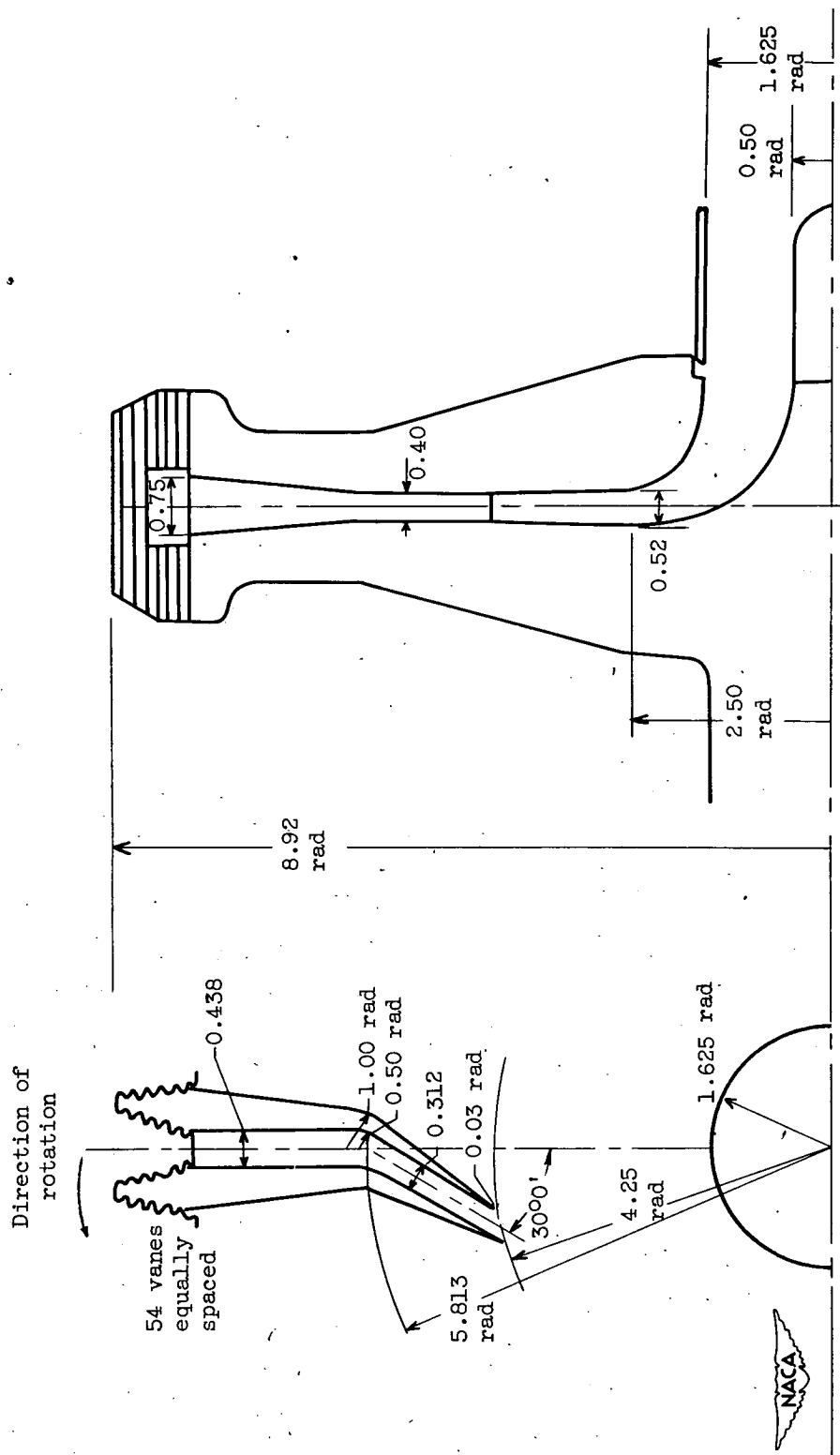
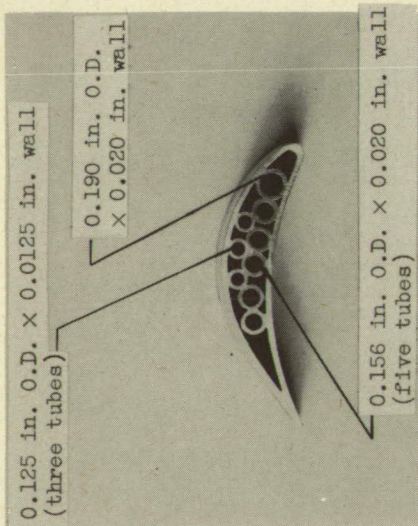
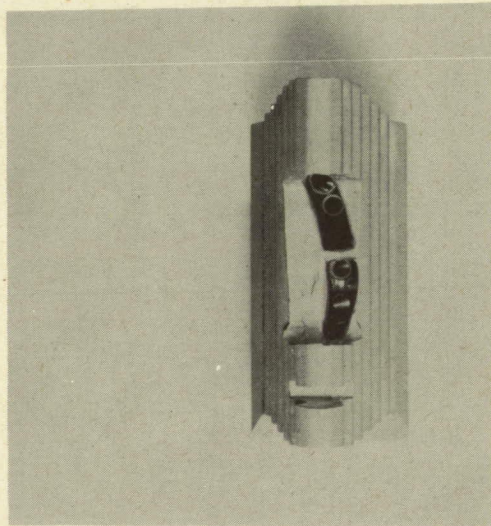


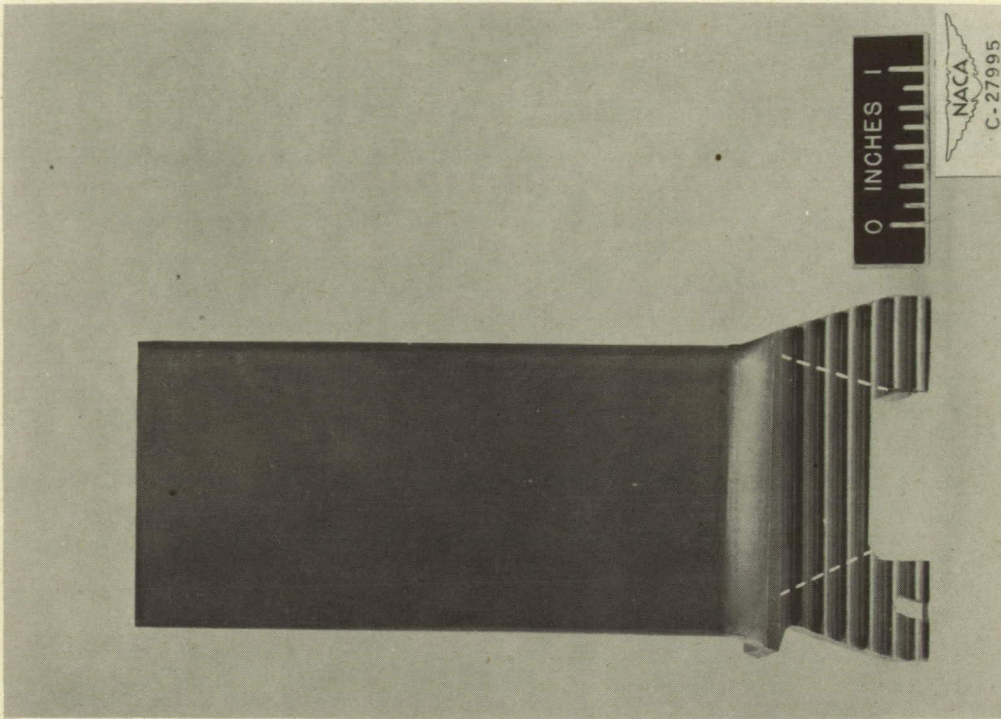
Figure 3. - Cooling-air impeller. (All dimensions are in inches.)



(a) Tip profile showing tube arrangement and size.



(b) Coolant passage entrance in base.



(c) Side view.

Figure 4. - Cast X-40 alloy air-cooled blade with nine-tube insert used in J33 turbine rotor.

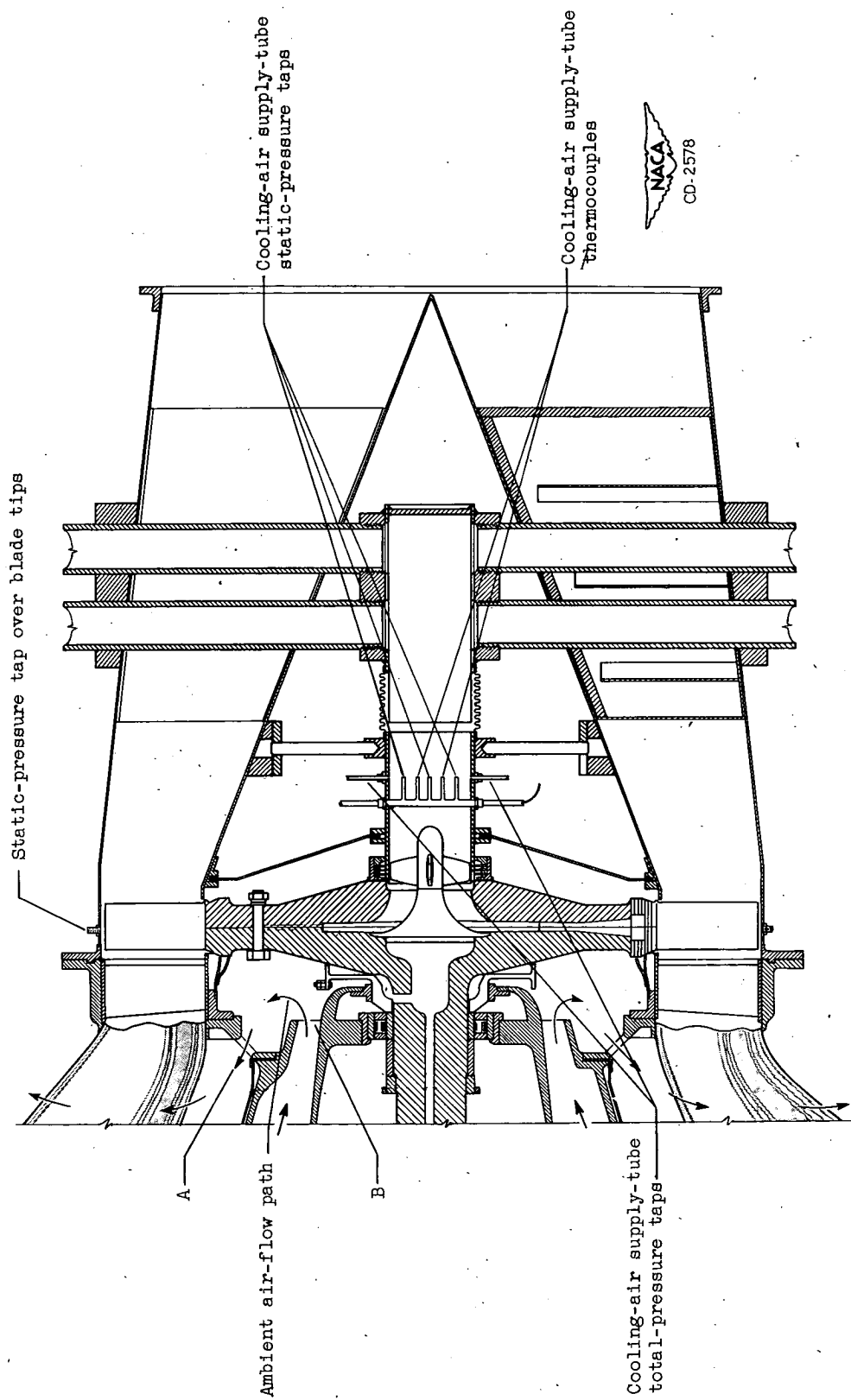


Figure 5. - Schematic sketch showing engine modification and cooling-air instrumentation.

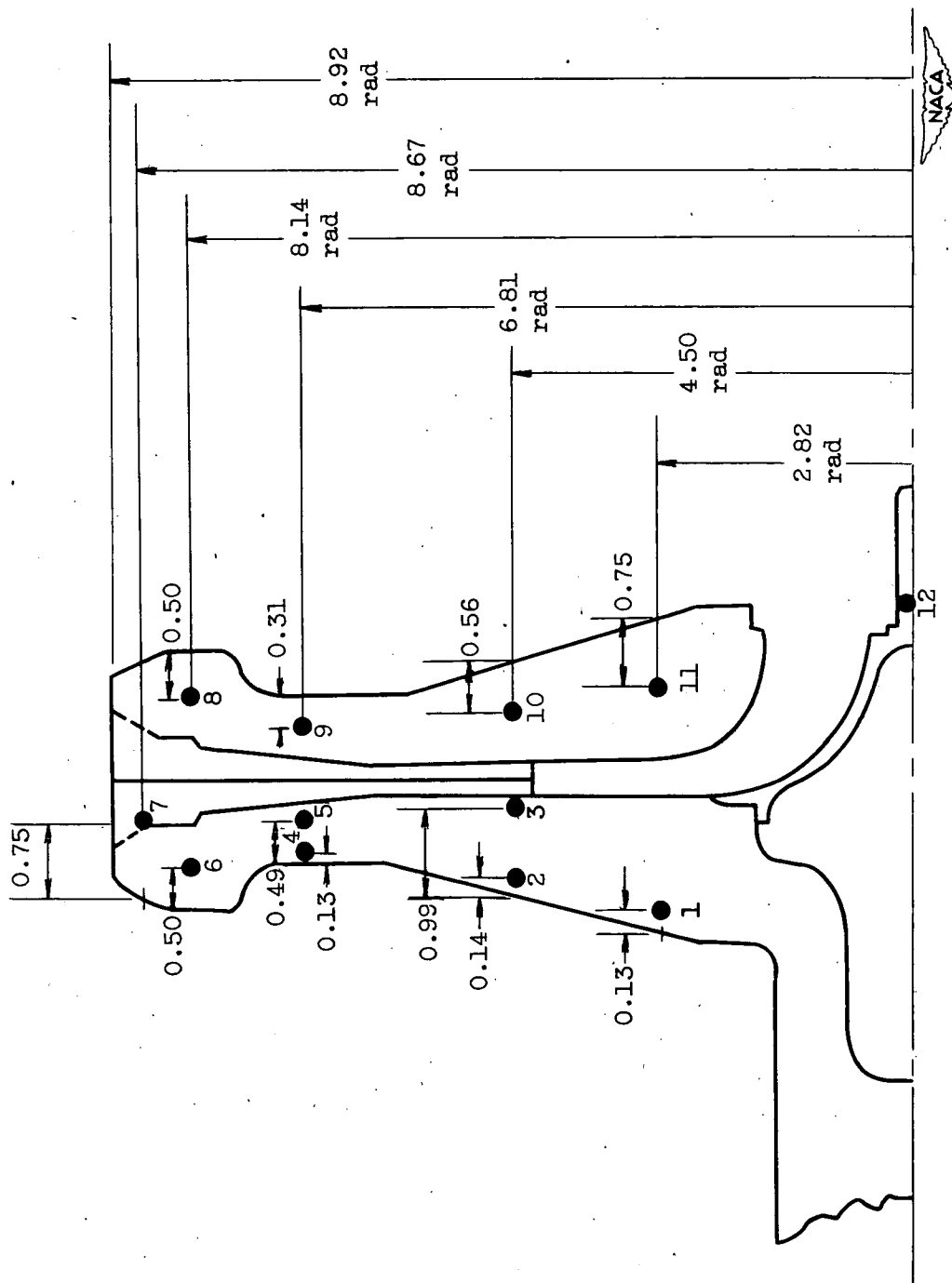


Figure 6. - Turbine disk thermocouple locations. (All dimensions are in inches.)

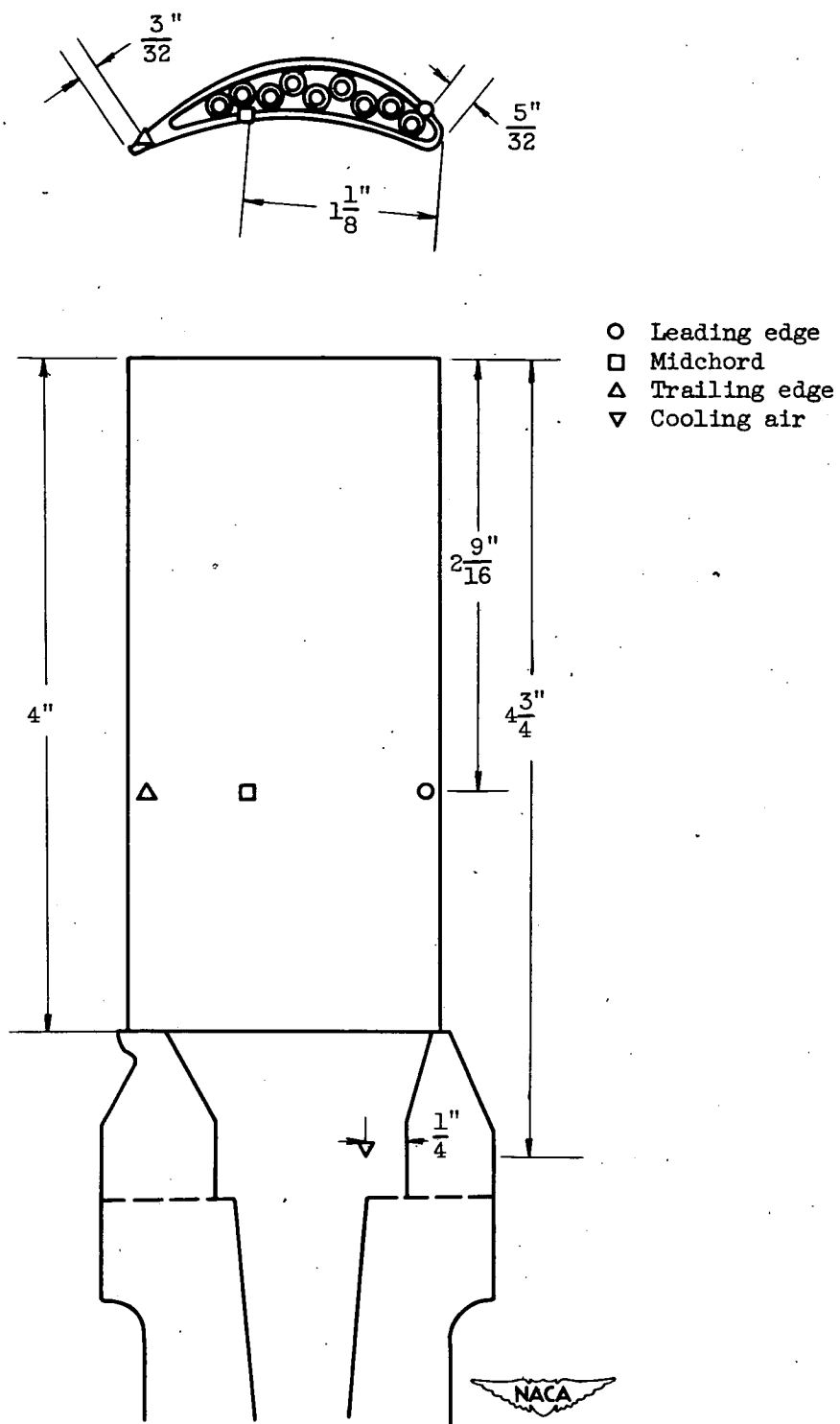


Figure 7. - Location of thermocouples in instrumented blades.

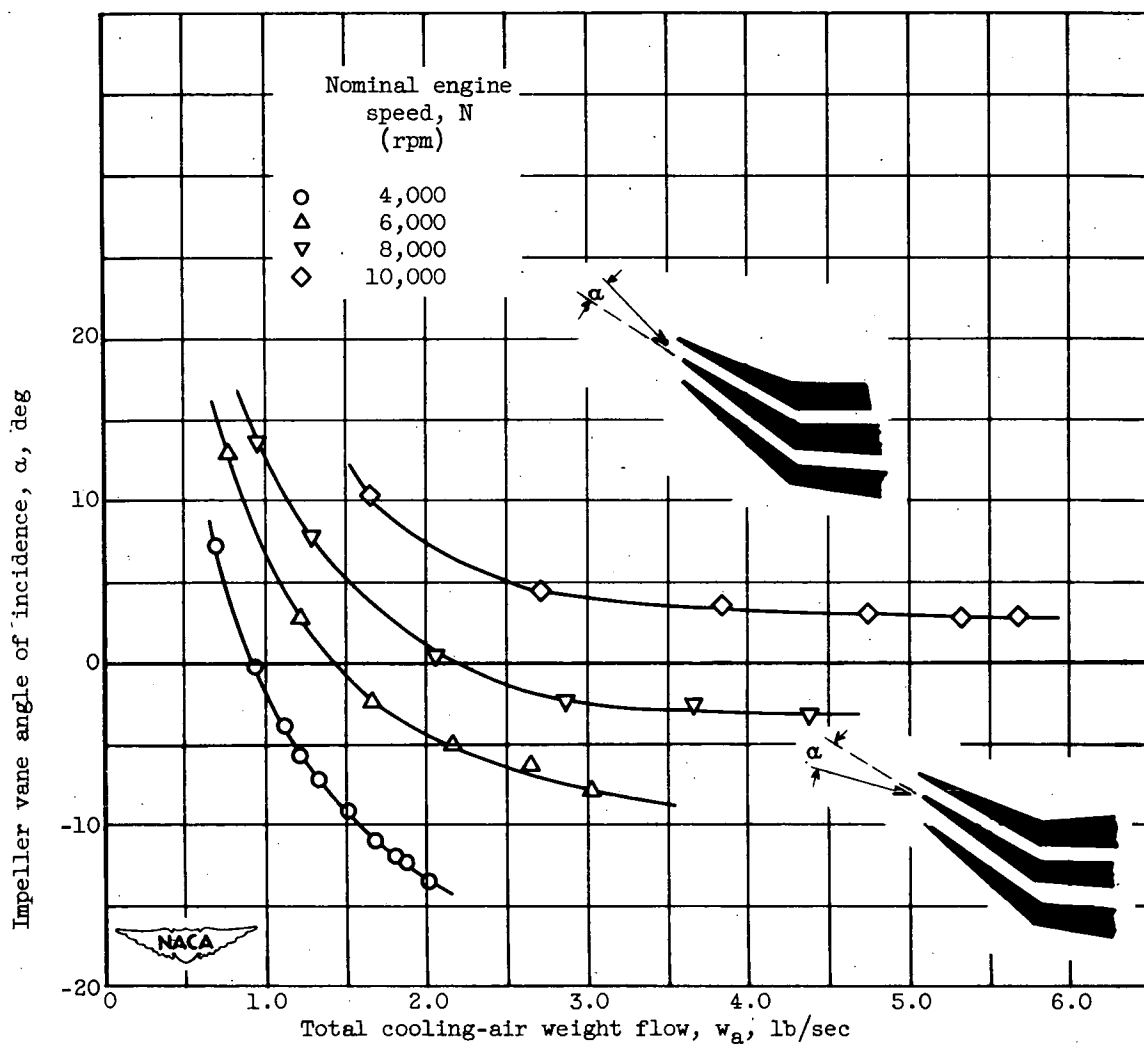


Figure 8. - Variation of impeller vane angle of incidence with cooling-air weight flow for various engine speeds.

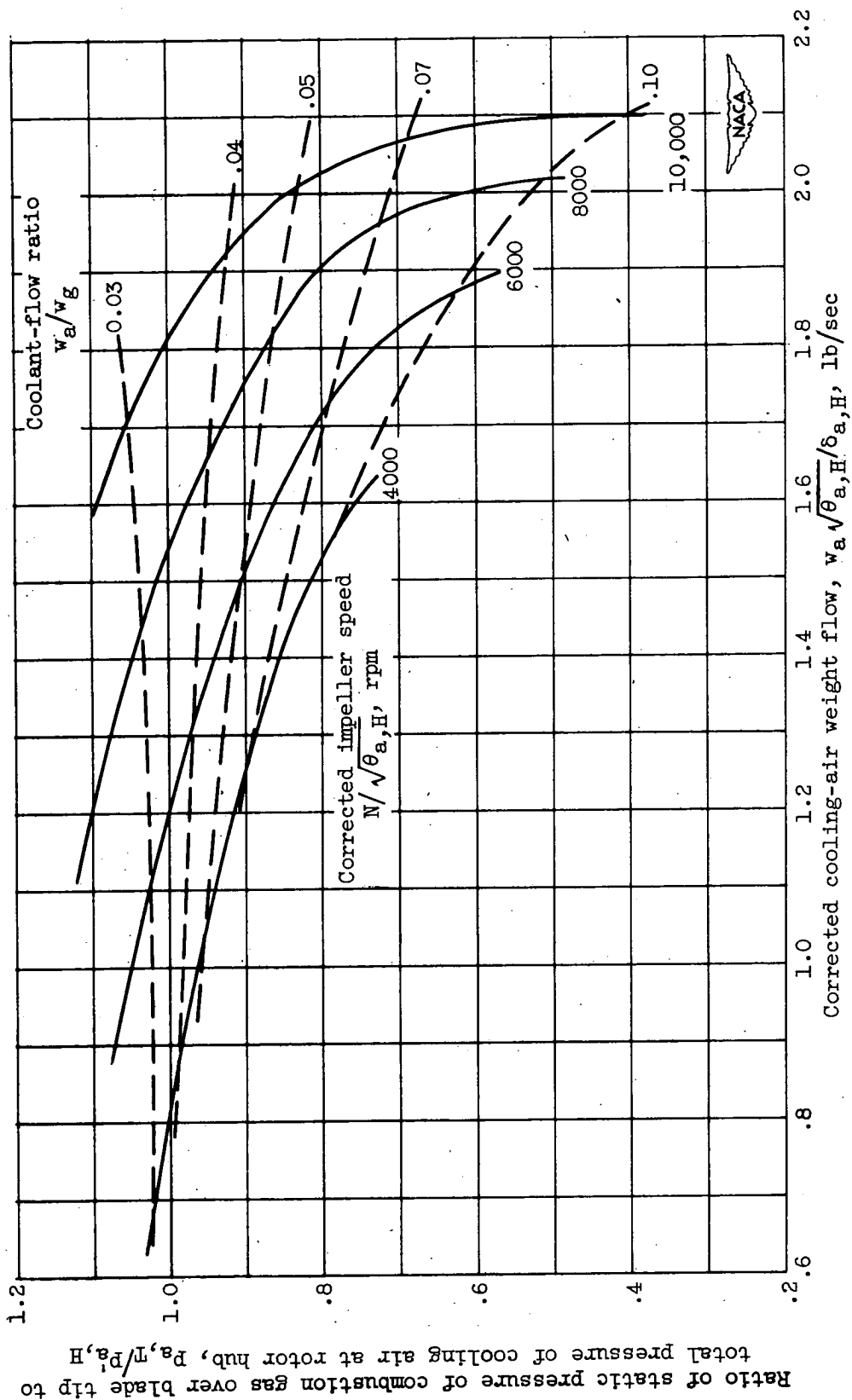


Figure 9. - Impeller cooling-air-flow characteristics.

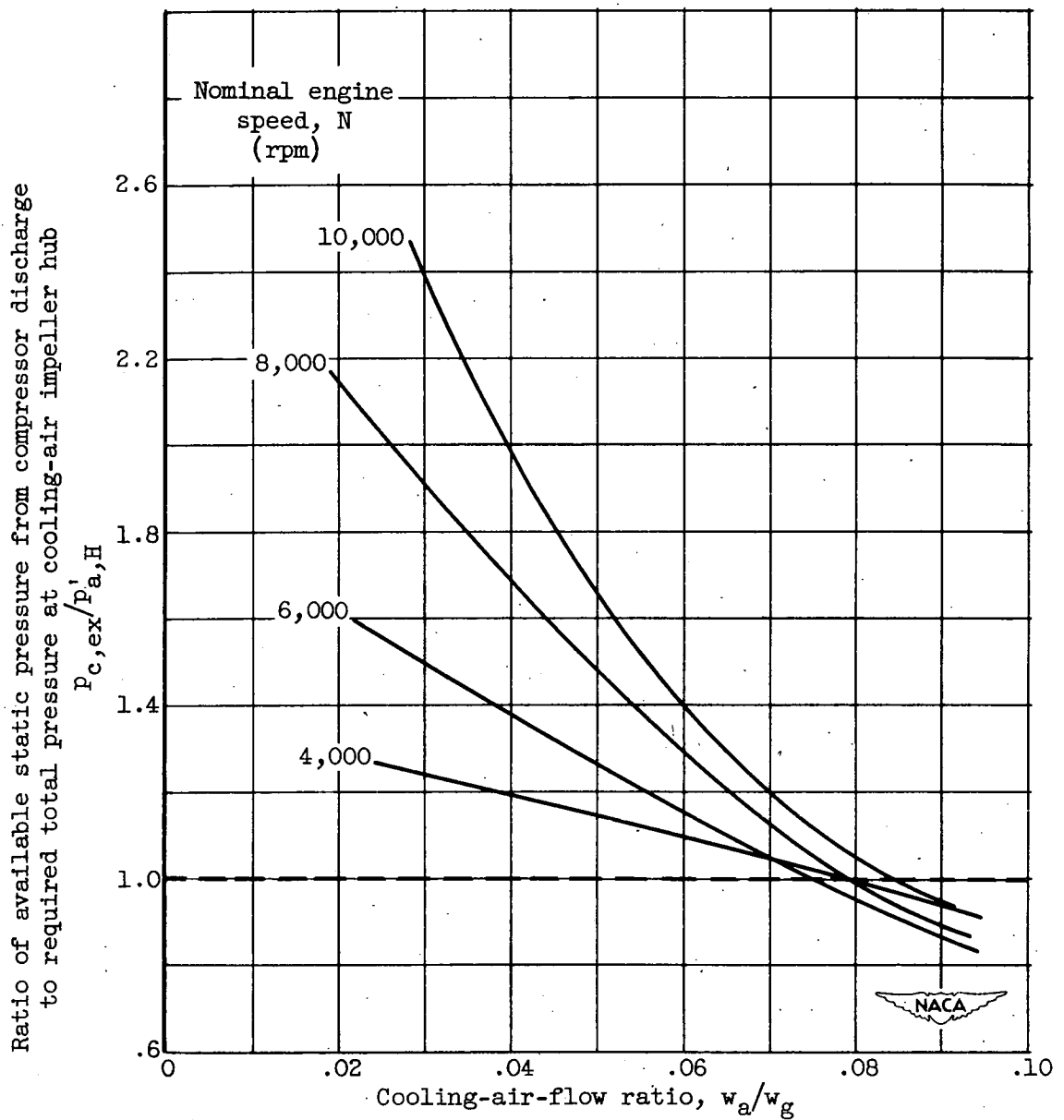
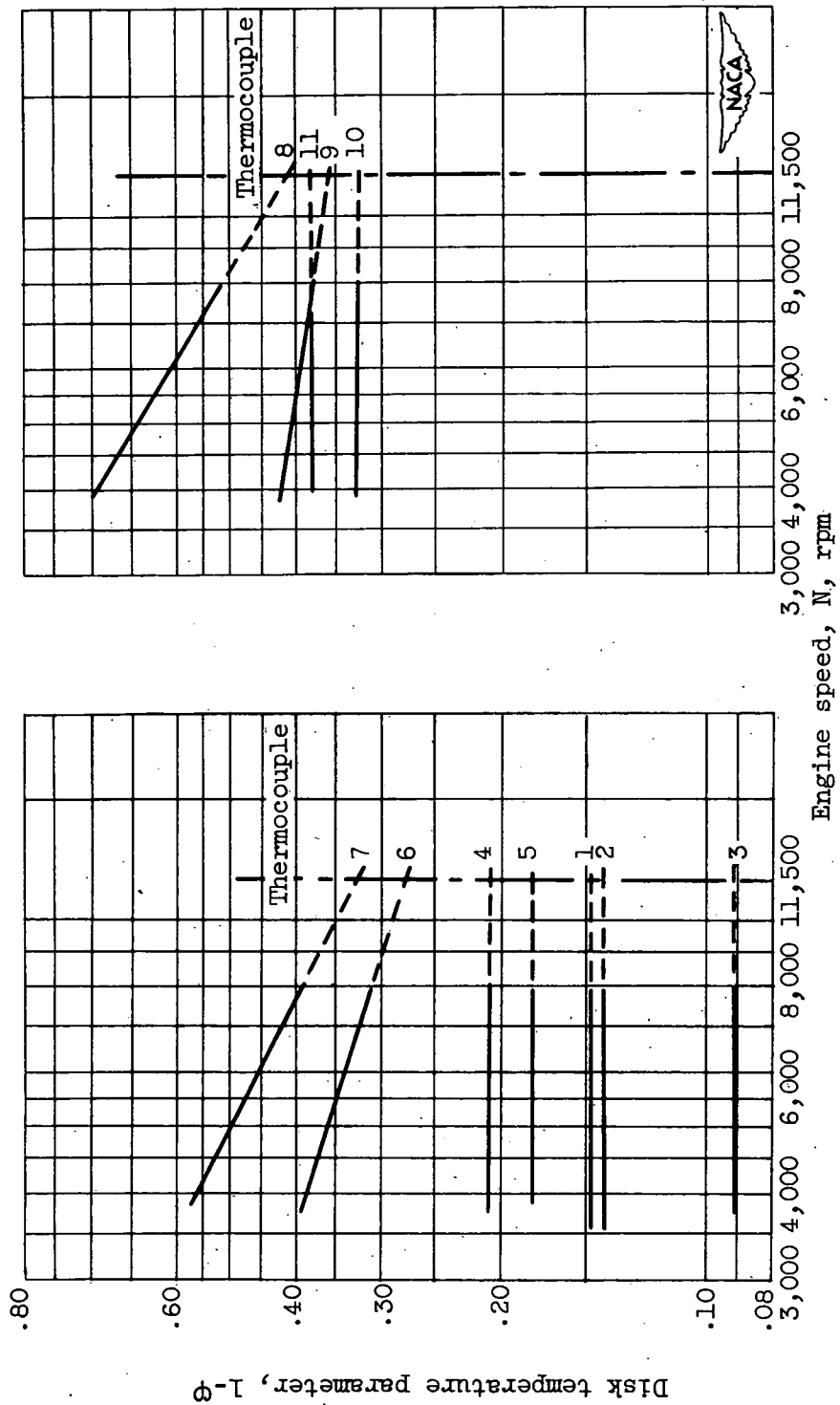


Figure 10. - Variation of ratio of available static pressure from compressor discharge to required total pressure at cooling-air impeller hub with cooling-air-flow ratio for various speeds.



(a) Forward disk.

(b) Rear disk.

Figure 11. - Correlation of disk temperature parameter $1-\phi$ with engine speed. Cooling-air-flow ratio, 0.02.

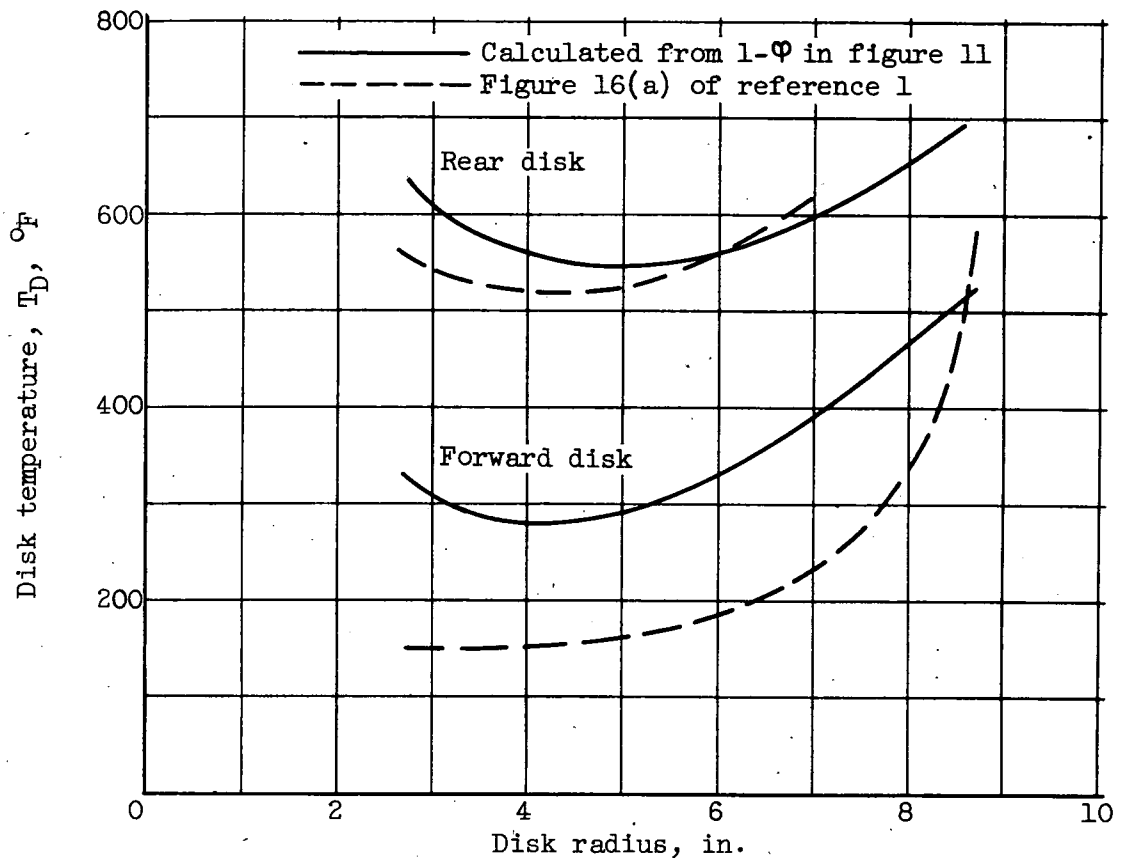


Figure 12. - Comparison of calculated disk temperatures for modified J33 engine. Engine speed (rated), 11,500 rpm; cooling-air inlet temperature, 120° F; cooling-air-flow ratio, 0.02; average effective gas temperature, 1445° F.

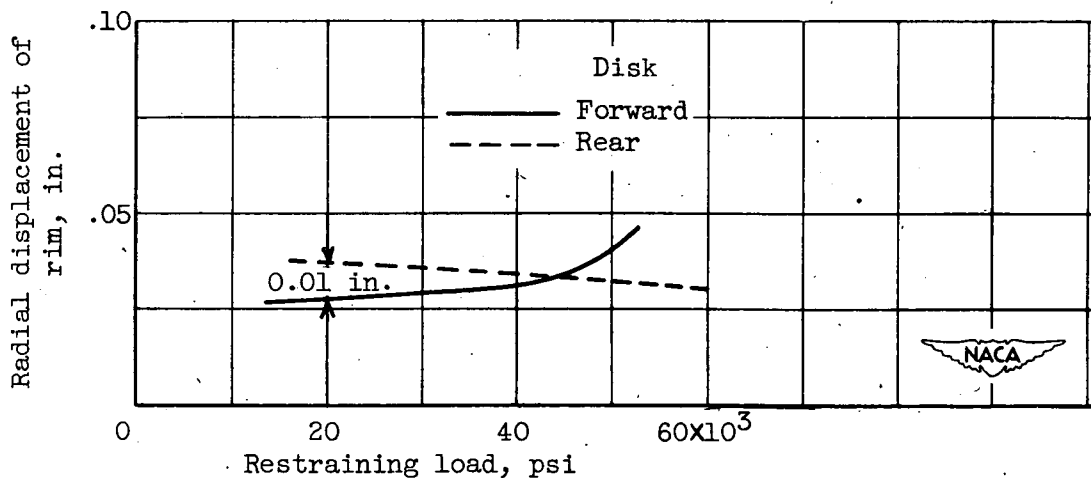


Figure 13. - Graphical determination of restraining loads at rated engine speed (11,500 rpm).

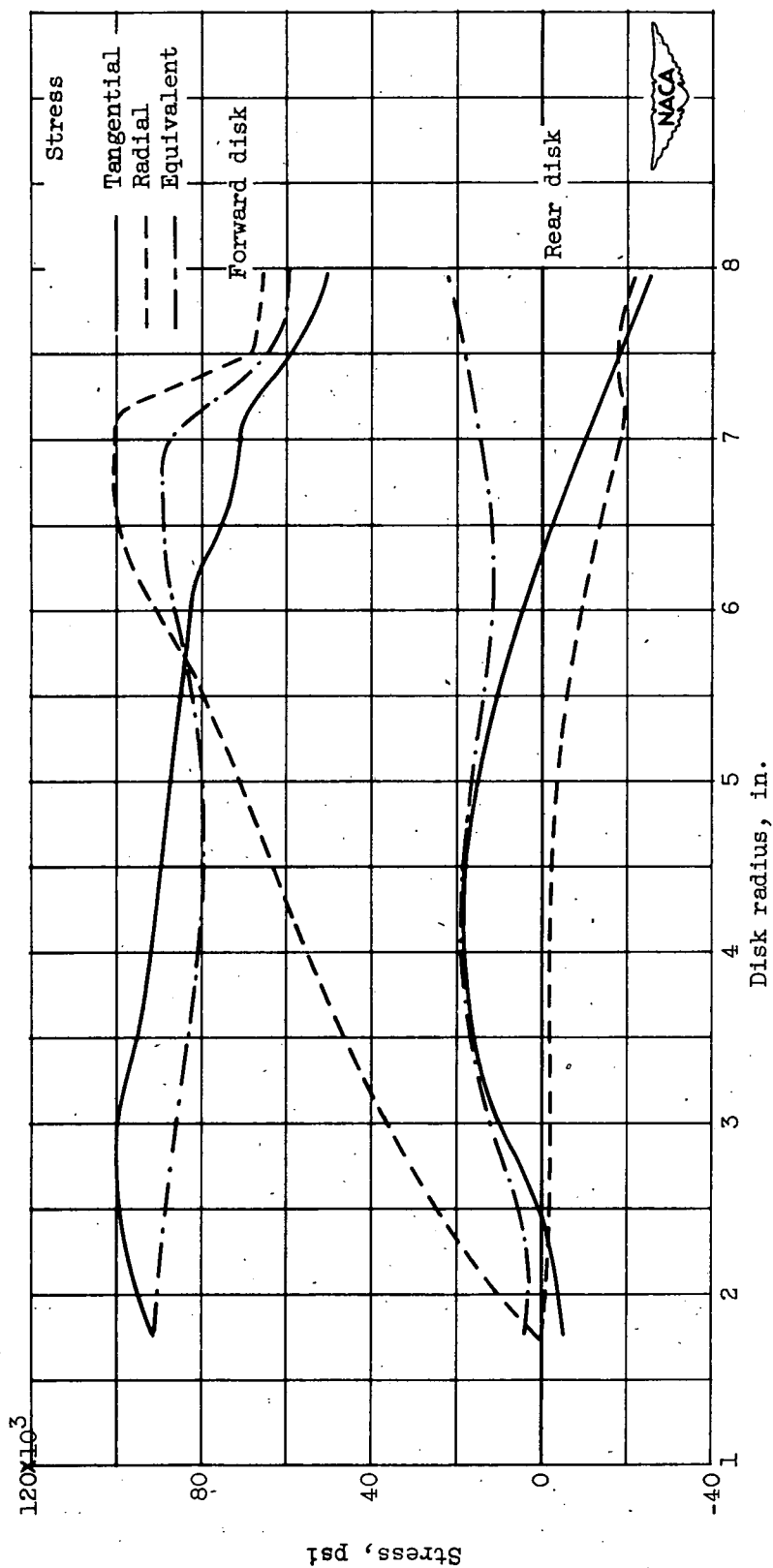


Figure 14. - Plastic stresses for forward and rear disks having dissimilar temperature distributions at rated engine speed (11,500 rpm).

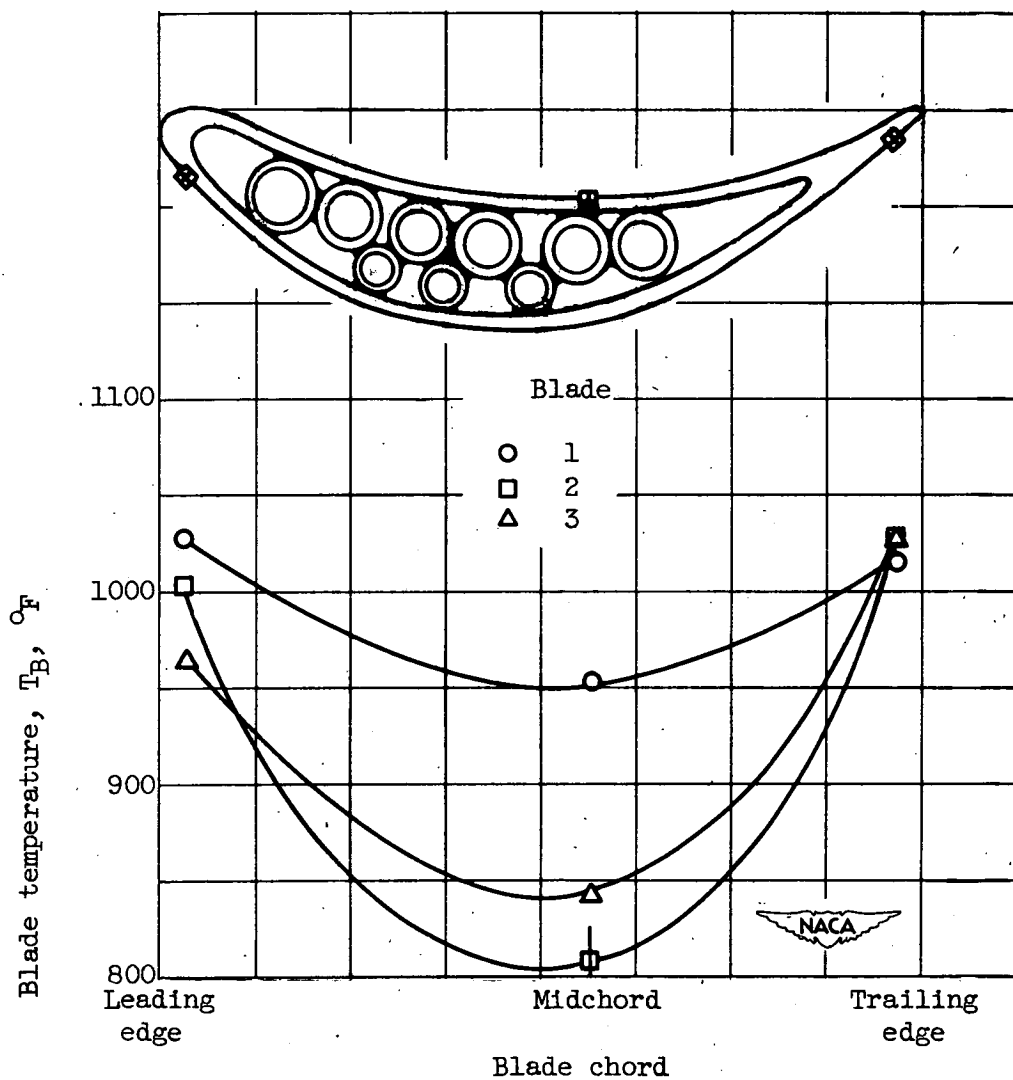


Figure 15. - Comparison of measured blade temperatures. Engine speed, 6000 rpm; cooling-air-flow ratio, 0.021; effective gas temperature, 1087° F; effective cooling-air temperature at blade base, 185° F.

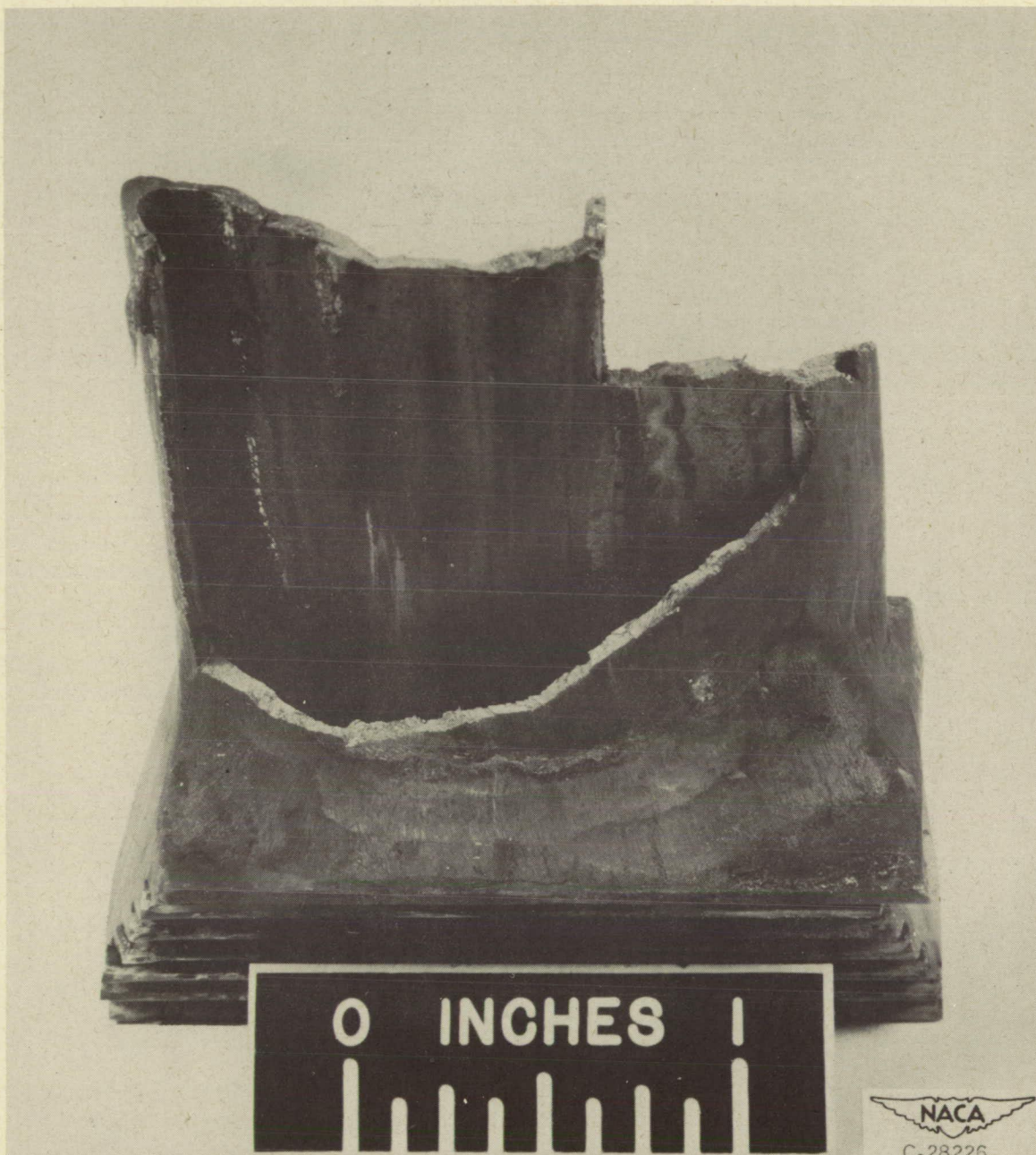


Figure 16. - Blade 1 after failure.

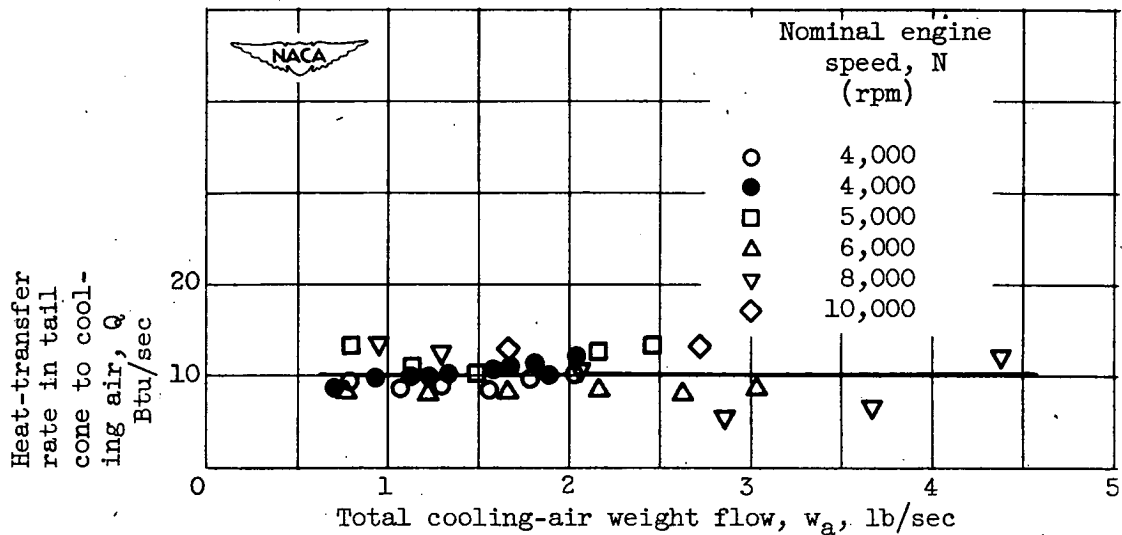


Figure 17. - Variation of heat transferred to cooling air while flowing through tail cone with cooling-air weight flow.

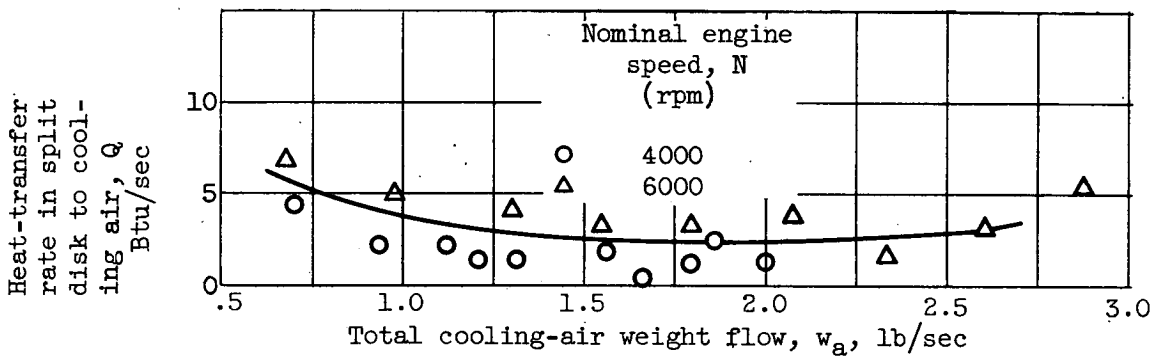


Figure 18. - Variation of heat transferred to cooling air from inside walls of split disk with cooling-air weight flow.

